POWER PLANT ENGINEERING

Trade Mark, Reg. U. S. Pat. Office

VOL. XXVII CHICAGO, DECEMBER 15, 1923 No. 24

Power Problems of Macy's Department Store

EXTENSIVE CHANGES MADE IN EQUIPMENT TO MEET REQUIREMENTS OF NEW ADDITION; OIL BURNING AND OTHER IMPROVEMENTS WILL LEAD TO LARGE CUT IN OPERATING EXPENSES

TO PROVIDE for the requirements of increased business, R. H. Macy & Co., New York City, is completing an addition to its department store building which will give an increase in cubic capacity of nearly 50 per cent. The original Macy store, fronting on Broadway, 34th and 35th streets, covers a ground area of 90,500 sq. ft., is 10 stories in height, and has a cubic capacity of approximately 16,000,000 cu. ft. The new addition lies directly to the west fronting on 34th and 35th streets, has a ground area of 26,600 sq. ft., and is 19 stories high. Including two underground stories used exclusively for loading delivery strucks and for recharging storage batteries for these trucks, it comprises with the original building nearly 24,000,000 cu. ft.

The older building will now be given over primarily to the sales departments, only the first, third, fourth, fifth, sixth and seventh floors of the new building to the west being devoted to sales. The remaining floors of the addition are for special departments. The mezzanine over the first floor, part of the second floor not required by the employment department, and the mezzanine over this floor, will be devoted to locker space. On the eighth floor will be the cafeteria, also recreation rooms for the women employes, and library. The corresponding floor of the older building is taken up by the restaurant and kitchen. Reserve stock rooms occupy the ninth, tenth, and eleventh floors of the west building; the twelfth floor is given over to the controller's department; the thirteenth, to executive offices, library, and council rooms; and the fourteenth, to bureau investigation and advertising. On the fifteenth floor are buyers' offices; on the sixteenth, educational auditorium, classrooms, and instructors' offices; and on the seventeenth, hospital and special workrooms. The eighteenth floor is unassigned, and the nineteenth is occupied by the laundry. Above this level are pent houses containing tanks, elevator machinery, and exhaust fans. Similar machinery is located over the tenth floor level in the older building.

Steam and power are utilized in a variety of ways, including lighting; heating and ventilating; hydraulic and electric elevators; refrigerating system for cooling drinking water, restaurant refrigerators, and manufacture of ice cream; pneumatic cash system; laundry; compressed air system; and vacuum cleaner system.

So far as possible, the power and mechanical equipment is concentrated in the sub-basement of the original building, with the boiler and engine rooms and the refrigerating plant on the 34th Street side. The hydraulic elevator machinery and the equipment for the pneumatic cash system are also located at this point.

Practically every feature of the equipment has undergone extensive alteration or additions, and since no interruption of service could be permitted the job has had much of interest and difficulty. Among the changes incident to the work are: replacement of coal by oil as the fuel for the year round, retaining, however, enough boilers on coal firing to take care of the entire load if emergency should demand it; installation of 240-v. system for lighting and power supply, and consequent changes in the old 110-v. system to accommodate it to the new conditions; introduction of electric instead of hydraulic elevators in the new building, and substitution of motor driven centrifugal pumps for hydraulic elevator service in the original building in place of cumbersome steam driven units; elimination of two small main generating units and installation of two units of 750 and 250 kw. capacity, respectively, increasing the total rated capacity of main units from 2000 to 2600 kw.; and discarding of the old 60-T. absorption refrigerating plant in favor of an ammonia compression system of 110 T. capacity.

Changes in Steam Generating Plant

Coming to the boiler house, a peculiar condition existed, in that the original installation of B. & W. water-tube units was so far in excess of requirements that not only were no additional boilers required but it was actually permissible to keep five 250-hp. boilers on coal firing as a reserve in case of emergency, while the remaining six boilers—No. 1 of 290 hp., No. 2 of 430 hp., and Nos. 3 to 6 of 250 hp. each—were refitted for oil firing. The coal- and oil-fired boilers are ranged on opposite sides of an aisle, down the center of which runs a track for the delivery of coal and removal of ashes. The ashes are transferred in cans by lift to the Thirty-fourth street sidewalk, thence they are removed by truck. Coal, No. 2 buckwheat, is stored to the extent of 500 T., enough for about a week's supply; but since the oil burning equipment was put in operation in February last, there has been no occasion for its use.

Oil storage tanks, four in number, each having a capacity of 19,000 gal., are located in a row under the sidewalk, in the sub-basement of the west building. From these, the fuel is pumped through a 6 by 12-in., two-cylinder Worthington transfer oil pump geared to a 7½ hp. Westinghouse motor, to a service tank located in an enclosure adjacent to the boiler room. The capacity of this tank is 17,500 gal., giving a total available capacity of 93,500 gal. or about three weeks' to a month's supply. Mexican fuel oil with a heat value of about 18,300 B.t.u. per lb. and a specific gravity of approximately 16 deg. Baumé is employed.

This oil is too heavy for pumping at ordinary temperatures, and a primary heating system consisting of bent pipe coils in a box is provided in each tank. Whenever it is necessary to heat the oil in the tank hot water circulated by a motor driven centrifugal pump and heated by exhaust steam is forced through these coils and an oil temperature of 95 deg. F. may thus be obtained. Oil from the service tank is reheated to 240 deg. F. by pumping through the oil supply pumps into Griscom-Russell Reilly multicoil oil heaters before being delivered to the supply lines. Oil is received at the supply line at a pressure of 150 lb. gage, and 120 lb. is maintained on the return lines. The supply to individual burners is regulated by hand. Automatic regulators on steam to the supply pumps act to cut out these pumps immediately in the event of trouble on the supply lines. Each oil fired boiler is equipped with two Peabody oil burners, with the exception of the 430 hp. unit, which has three burners.

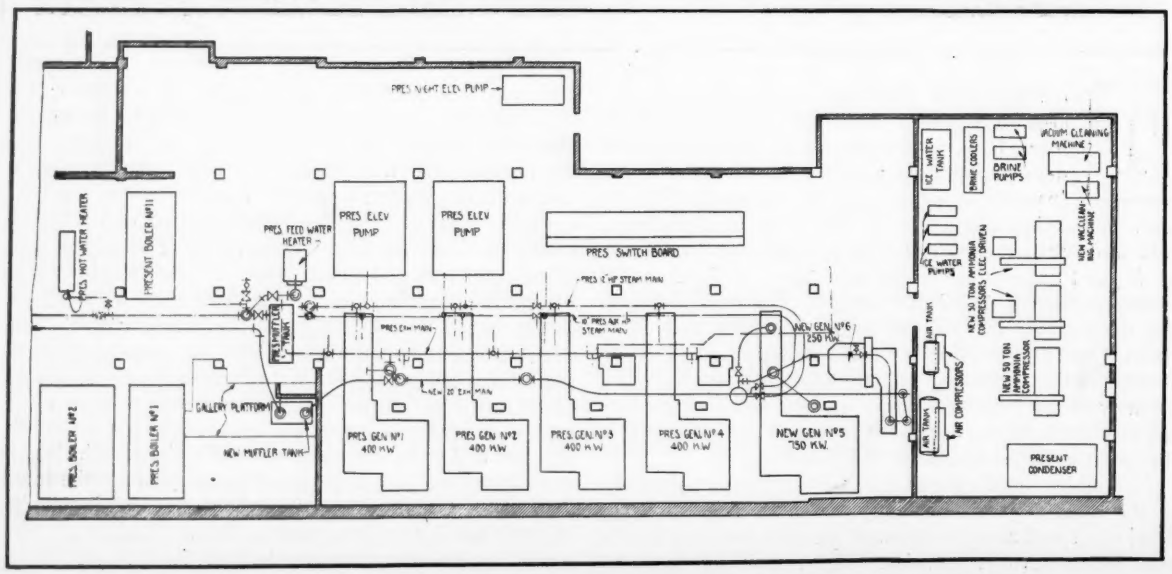


FIG. 1. ARRANGEMENT OF GENERATING EQUIPMENT AND REFRIGERATION APPARATUS

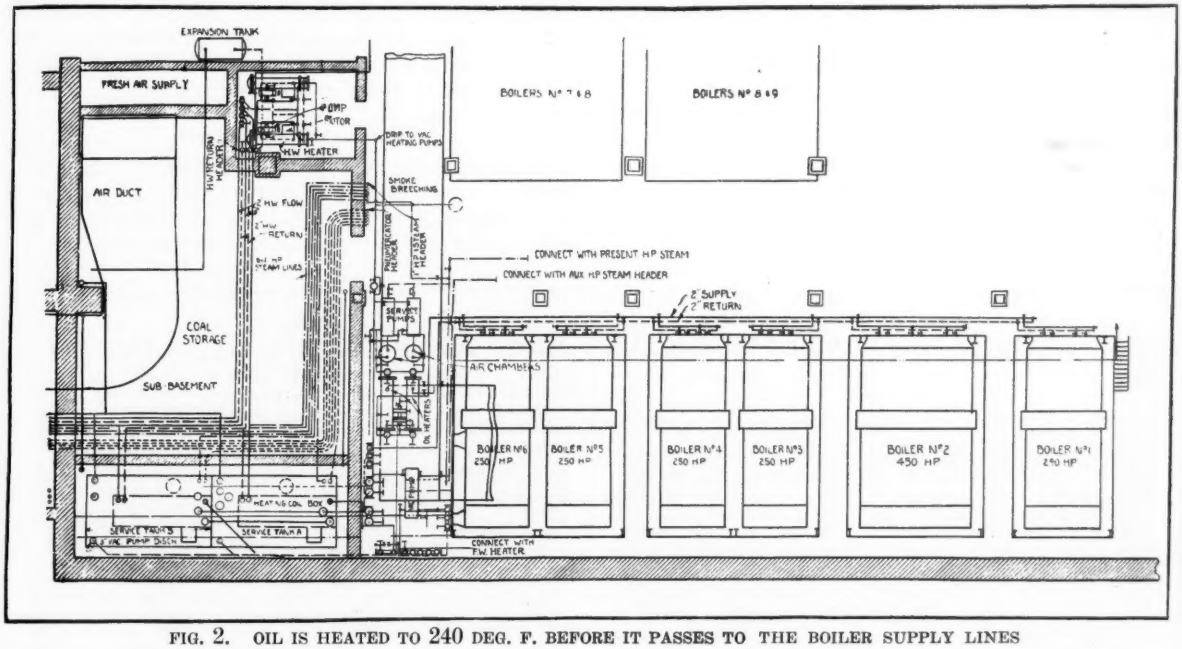


FIG. 2. OIL IS HEATED TO 240 DEG. F. BEFORE IT PASSES TO THE BOILER SUPPLY LINES

In the coal-fired boilers, the Parsons system of furnace construction and steam jets for forced draft is employed. Regulation of flue dampers is accomplished by hand, and is guided by reference to the Bailey flow meters recording steam and air flow and flue temperature and indicating furnace draft. One of these meters has been installed for each oil fired boiler.

In connection with the oil supply system a small vacuum pump is provided. This acts as a booster to draw oil from the tanks and prime the supply pumps in the emergency of their losing oil.

Oil consumption is checked by means of a pneumercator which is connected with the service tank and with each storage tank. In this way, an accurate daily reckoning may be had of the oil drawn from the service tank, with a check against the amount drawn from the storage tanks.

The boilers are equipped with pop safety valves of the Consolidated Safety Valve Co. The working pressure is 135 lb. gage. Main and auxiliary steam headers are provided, and the oil fired boilers are connected with both headers. The coal fired boilers are connected only with the main header. Diamond soot blowers are installed for frequent dusting of the outer tube surfaces.

Combustible waste from the store, amounting on the average to about 15 T. daily, has hitherto been consumed in an incinerator of inadequate size, without reclamation of waste heat. This condition is to be rectified by the installation of a new incinerator having a capacity of 1 T. per hour, with an economizer in the smoke connection. The resultant heat gain will be equivalent to 1000 T. of coal per year at a conservative estimate.

The two Knowles 10 by 16 by 8 by 10-in. compound duplex steam feed pumps are relocated, and will supply boiler feed water through the 2500-hp. Webster heater and the new economizer. The present system of steam traps is to be discarded for all boiler pressure drop returns, and will be replaced by the Holly loop gravity system.

Increase in Generator Capacity and Voltage

Original generating capacity of 2000 kw. was obtained with four Rice & Sargent tandem compound Corliss engines direct connected to 400-kw. Western Electric generators running at 100 r.p.m., and two small units of 200 kw. each. The latter have been removed, and two new units installed—one a Hamilton-Corliss tandem compound, 26 by 44 by 42 in., with 750-kw. Crocker-Wheeler generator running at 100 r.p.m., the other a 20 by 27-in. Erie-Ball simple Corliss engine with 250-kw. Crocker-Wheeler generator at 150 r.p.m. The change of voltage from 110 v. to 240 v. on the distributing system necessitated rewinding the old Western Electric generators for the new condition. Making allowance for the scrapping of two units, the net increase in rated capacity of the plant amounts to 600 kw.

In connection with change of voltage, the switchboard work involved considerable complication, as it was necessary to build a complete temporary switchboard behind the main board to handle the situation while the

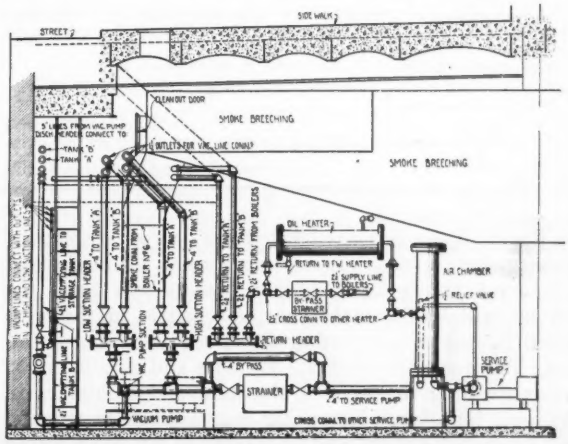


FIG. 3. PUMPING AND HEATING EQUIPMENT USED IN HANDLING THE OIL

latter was being enlarged and rewired. The west building has a three-wire system, while the original building retains its two-wire system with the new voltage. Balance is maintained on the two-wire system by splitting feeders and using three 500-amp. motor generator sets as balancers. One of the three normally takes care of the unbalanced load of automobile charging panels in the new building; but this unbalanced load may be used interchangeably with that in the old building for balancing the system.

Prior to the building of the addition, the power generated amounted to approximately 4,000,000 kw.-hr. per year. With the new conditions, this will be increased to about 7,000,000 kw.-hr. per year.

Applications of Power

Aside from the changes in the generating plant, considerable economies will be introduced by alterations in the refrigerating and hydraulic elevator systems, as well as by improved equipment in the new building.

Mechanical Equipment in the Macy Store After Changes Were Made

GENERAL

Building: 19 story addition of nearly 8,000,000 cu. ft. capacity to original building of 10 stories and 16,000,000 cu. ft.
Architect: Robert D. Kohn, New York.
Builders: Marc Eidlitz & Son, New York.
Consulting engineer for steam power, heating and ventilating, plumbing, and refrigerating equipment: Werner Nygren, New York.
Consulting engineer for electric and elevator equipment: Kaiser, Muller & Davies, New York.
Contractor for steam power, heating, ventilating and oil burning equipment: Gillis & Geoghegan, New York.

BOILERS

B. & W. water tube. Six with aggregate rated capacity of 1720 hp., oil-fired; five with 1250 hp., coal fired.

OIL BURNING EQUIPMENT

Burners.....Peabody Engineering Corp.
4 storage tanks, 19,000 gal. each, and 1 service tank, 17,500 gal., built by Dover Iron Co.
Oil heating equipment.....Reilly heaters, Griscom-Russell Co., one for hot water circulating to oil tanks, and two for oil heating before delivery to supply lines.
Oil transfer pump.....Worthington 6 by 12 in., two-cylinder, geared to 7½-hp. Westinghouse motor.
Oil service pumps.....2—7 by 3½ by 8-in. Worthington, with regulators to cut out if trouble occurs on supply line, and a small vacuum pump for priming if pumps lose oil.

MAIN UNITS

4 old Rice & Sargent tandem compound Corliss engines, Providence Engineering Wks., with 400 kw. Western Electric generators 250 v., 100 r.p.m.
1—26 by 44 by 42-in. Hoover, Owens, Rentschler Hamilton-Corliss tandem compound engine with 750 kw. Crocker-Wheeler generator, 250 v., 150 r.p.m.
1—20 by 27 in. simple Erie-Ball Corliss engine, with 250 kw. Crocker-Wheeler generator, 250 v., 150 r.p.m.

REFRIGERATING SYSTEM

1—50 T. Voss ammonia compressor, 12 by 24 in., driven from 100 hp. type C compound wound Sprague motor, 417/625 r.p.m.
2—30 T. compressors of same make, driven from 60 hp. Sprague motors, 312/625 r.p.m.
1 condenser, 2-pipe, 9 stacks, 18 ft. long, 10 pipes high.
2 brine coolers, shell and tube type, 750 sq. ft. each.
Water cooling tank.....1—14 by 7 by 7 ft.
Brine pumps.....Blake-Knowles horizontal duplex, 1—10 by 6 by 12 in. and 1—10 by 6 by 10 in.
Cold drinking water pumps.....Worthington duplex double-acting, 1—7½ by 4½ by 10 in. and 2—10 by 4½ by 10 in.
Ammonia piping.....All ex. hvy. W. I. with ex. hvy. steel fittings.
Regulation.....General Electric hand control.

HYDRAULIC ELEVATOR PUMPS

Worthington centrifugal hydraulic elevator service pumps..... .....1—2200 gal./min., 2—1500 gal./min. and 1—400 gal./min., driven by 350-hp., 200-hp., and 75-hp. motors, respectively.

ELEVATORS

23 hydraulic elevators and 2 sidewalk lifts; 7 electric esculators, 1 lowerator, and 1 dumbwaiter in original building.
17 electric elevators 16 esculators, and lift conveyor in west building.

HEATING AND VENTILATING SYSTEM

12 supply fans and 11 exhaust fans, ranging in power from 15 hp. to 110 hp.
Vento stacks on 11th floor of west building.
24 in. Cochrane multiport back pressure valve in exhaust main at 19th floor, and three branches from exhaust main to supply heating and ventilating system.

MISCELLANEOUS

Feed pumps.....2—Knowles 10 by 16 by 8 by 10 in. compound duplex steam pumps.
Drip system.....Holly loop gravity return
Feed water heater..... 2500-hp. Webster-Starr feed water heater, filter, and purifier.
Economizer.....Type to be determined.
Safety valves.....Pop valves, Consolidated
Blowoff valves.....Pratt & Cady plug cocks with angle valves.
Soot blowers.....Diamond.
Boiler meters.....Bailey.
High pressure steam piping.....Ex. hvy. W. I. up to 4 in. in size, steel pipe with Van Stone joints for piping 5 in. or larger.
Oil piping.....Ex. hvy. W. I.
Feed water piping.....Brass, with auxiliary feed of W. I.
Exhaust piping.....Std. W. I. up to 12 in. in size and std. steel with Van Stone joints for piping 12 in. and larger.
Crane.....10 T. Reading Crane & Hoist Works.
Oil measurement.....Pneumercator.
Rotary blowers for cash system..... .....2—Abraham Green Blower Co., driven through chain belts from 65/94-hp. Crocker-Wheeler motors.
Air compressors.....2 Fairhurst 10¼ by 8 in. and 10¼ by 10 in., driven through belts from 30 and 40 hp. motors.
Sundh system of electrically controlled pressure regulators.
Vacuum cleaning machines, pumps for house service, fire pump, etc.

Pedestal switches are located beside each generator, with control from the main switchboard where the circuit breakers are fixed. Main feeders run from the switchboard to the main charging panels in the west building, and to distributing switchboards in that building. There are also separate switches and circuit breakers controlling the electric elevators and esculators in the west building. Recording wattmeters for measuring generated output and consumption by distributing feeders are also mounted on the main board.

The old 60-T. absorption refrigerating plant has been removed, and replaced by an ammonia compression system of 110 T. capacity. There are three Voss ammonia compressors, one of 50 T. and two of 30 T., driven respectively by 100-hp. and 60-hp. Sprague motors. Work of refrigeration is done in a drinking water cooling tank 14 by 7 by 7 ft. and in two shell and tube brine coolers 13 ft. long having 750 sq. ft. of cooling surface each. The brine system is of the balanced type, with a balancing tank located above the tenth floor of the old building.

The condenser consists of nine stacks of two-pipe condensing coils 18 ft. long and 10 pipes high. For circulating cold drinking water and brine, as required, three Worthington duplex cold water pumps and two Blake-Knowles horizontal duplex double-acting brine pumps are employed.

There will be four new elevator pumps, all of centrifugal type and motor driven, as follows:

2—1500 gal. per min.....200 hp.
1—2200 gal. per min.....350 hp.
1— 400 gal per min..... 75 hp.

They will take the place of two 22½ by 28-in.—28 by 14¾ by 24-in. Laidlaw-Dunn-Gordon hydraulic steam pumps and one smaller reciprocating steam pump which is being used for night elevator service. After allowing for the space required by the automatic control equipment, there will be a saving in floor room of about 400 sq. ft. from this change; but the chief benefit will, of course, be derived from increased economy of operation and maintenance.

FIG. 4. TWO NEW GENERATING UNITS WERE INSTALLED AND THE BOILERS CHANGED TO OIL FIRING

On the left are shown the new generating units, the smaller one rated at 250 kw. and the larger one at 750 kw. In the view on the right, one of the six oil fired boilers is shown. At the right of this boiler are the oil supply pumps with the heating tanks above.

Two Green rotary pressure blowers, chain driven from variable speed Crocker-Wheeler motors with automatic control, create the vacuum for the cash system. There will also be two Fairhurst air compressors, driven through belts and cranks from motors of 30 and 40 hp. respectively, for supplying compressed air to operate elevator doors and to meet other requirements about the building. Vacuum cleaning machines and a number of small pumps for various classes of house service complete the apparatus in the boiler and engine rooms.

Automatic control of the Fairhurst air compressors will be exercised through the Sundh system of electrically operated pressure regulators, and a switchboard for this purpose is being installed by the Sundh Electric Co., of Newark. A similar but much more complicated switchboard will be required for the automatic regulation of the new hydraulic elevator pumps, which will also be controlled by the Sundh system.

In the old building are 23 hydraulic elevators, two sidewalk lifts, seven esculators, a lowerator, and a dumbwaiter. All the elevator equipment in the west building is electrically operated, including 17 elevators, two from the basement to the first floor, 16 esculators, and a parcel conveyor.

Above the tenth floor of the older building are six fans for fresh air supply, and two exhaust fans serving the kitchen and restaurant. Two more supply fans are located on the tenth floor of this building, and two others in the basement deliver heated air to the entrances on Thirty-fourth and Thirty-fifth streets. Two others, yet to be installed, will serve the engine room.

A 110-hp. exhaust fan on the nineteenth floor of the west building serves the two sub-basement floors where loading and charging of trucks is done. Four others are located in the pent houses above this floor, two of which are for general exhaust from the old building and the lower stories of the addition, one for the upper stories of the addition, and one for the laundry. A small exhaust fan is used for the motion picture auditorium for employes' training; and two fans in the cellar are primarily intended for exhaust from the engine and boiler rooms.

Motor sizes on these fans run from 15 hp. to 110 hp. The 12 supply fans have a total motor horsepower of 500, and deliver 907,000 cu. ft. of air per min.; while 11 exhaust fans show a total of 480 hp., with capacity for removal of 855,000 cu. ft. per min.

A 24-in. Cochrane multiport back pressure valve is set in the exhaust main at the ceiling of the nineteenth story of the west building. Just back of this point, a branch is taken off, which connects with all the risers for heating the new building. At the ceiling of the ninth floor, a distributing main branches from the exhaust main and connects with all the heating risers of the older building in order to re-enforce them. A third branch of 20-in. diameter is taken off at the ceiling of the tenth floor and supplies the various "Vento" tempering stacks for warming fresh air for ventilation.

Steam is also employed for various other purposes, as in the laundry and for cooking in the kitchen.

Records and Economic Results

Installation of recording instruments is as yet incomplete, and for that reason no figures on evaporation are

available. The Bailey boiler meters on each boiler furnish a check on boiler room operation. Oil delivery from the tanks and supply to the burners are checked against each other by the pneumercator. At the switchboard, wattmeter readings show the energy output of the main units and its distribution to the feeders.

It is far too early to present definite figures as to savings accomplished, since the installation is still incomplete; but there is no doubt that the final economic balance will make a good showing. In one feature, the results of change are already clear. The introduction of oil burning has reduced the item of labor at this plant by $15,000 a year; and as the system has been in operation since February, we are able through the courtesy of the operating engineers to present some reliable figures on the gain from oil burning. The accompanying tabulation summarizes certain items for the months of May, 1922, and May, 1923. Owing to the unsettled condition of the coal market in the summer and fall of 1922. a fair comparison could not be had on the basis of coal costs during these months.

It appears that the saving in the fuel item alone, with load conditions unchanged, is approximately $2000 a month, while the saving of $1250 a month in fire-room labor brings the total well over $3000 a month. The increase in power consumption with the west building in operation will bring a corresponding increase in this figure, which will then exceed $50,000 a year due to the change to oil fuel. With the further economies due to utilization of heat from combustible waste, substitution of electric for steam drive, increase in voltage, etc., other substantial savings will be apparent.

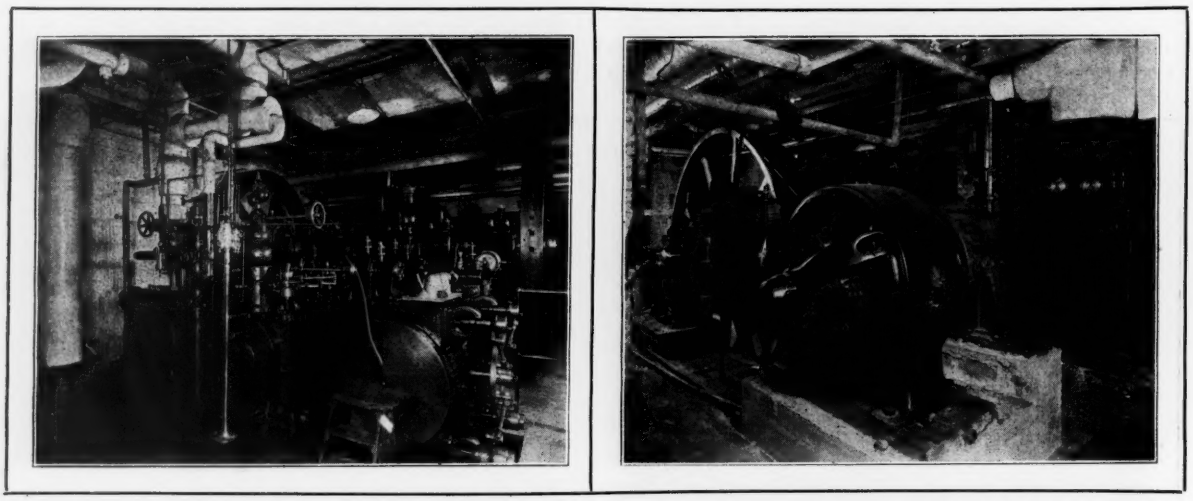


FIG. 5. IMPROVEMENTS HAVE BEEN MADE IN THE ELEVATOR AND REFRIGERATION EQUIPMENT

At the left is shown one of the two steam driven hydraulic elevator pumps which have been discarded in favor of electric driven centrifugal pumps with considerable saving in operating costs and reduction in floor space.

At the right is the 50-T. ammonia compressor which is belt-driven by a motor. The control board shown is the Sundh control for the Fairhurst compressors which supply air for elevator doors and other service.

OPERATING COSTS WITH COAL AND OIL COMPARED

	May, 1922	May, 1923
Fuel consumed	2,863,667 lb.	188,325 gal.
Kw.-hr. generated	338,220	324,758
Lb. fuel per kw-hr.	8.47	4.6437
Average daily consumption	92,376 lb.	6,075 gal.
Estimated consumption by steam driven units:		
Elevator pumps	252 T.	31,611 gal.
Restaurant, sump ejectors, Kenny machines, etc.	115 T.	14,442 gal.
Ice plant	60 T.	7,530 gal.
	427 T.	53,563 gal.
Lb. fuel per kw.-hr. consumed by generating plant	5.94	3.3230
Cost of lubricating oil, waste, etc.	$458.33	$660.00
Cost per kw.-hr. of all fuel, waste, lubricating oil, etc., and salaries of men employed in generating plant, added together, cents	2.858	2.2726
Gross saving in fuel alone, without allowances for steam used outside generating plant,		

In closing, acknowledgment is made to Werner Nygren, consulting engineer on the steam power, heating and ventilating, plumbing, and refrigerating equipment, for the co-operation necessary in preparing this article.

THROUGH THE Bureau of Mines, the Department of the Interior has recently issued a paper on the characteristics of and burning fuel oils as the third of a series of reports in a survey of petroleum products manufactured on the Pacific Coast from California crude oils. California crudes differ in many characteristics from the petroleum produced in the other parts of the United States, and the claim has been made that some of the present Federal specifications for petroleum products tend to discriminate against California products. The purposes of the present survey are to determine what particular specification requirements California petroleum products characteristically fail to meet, how widely the products deviate from these requirements, and if possible to show whether the fault lies with the specifications or the products, or along what lines investigation to determine this question should proceed. The first report of the series gave the physical and chemical properties of the samples of gasoline and mineral spirits. A second report gave the physical and chemical properties of samples of lubricating oils. Copies of Serial 2511, "Survey of Pacific Coast Petroleum Products, Part 3, Burning and Fuel Oils," by Earl C. Lane and N. F. LeJeune, assistant chemists, may be obtained from the Department of the Interior, Bureau of Mines, Washington, D. C.

Correcting Trouble in Small Steam Turbines

PROMPT ATTENTION TO FIRST SIGNS OF TROUBLE WILL OFTEN PREVENT COSTLY REPAIRS IN THE END. BY R. A. CULTRA

SMALL STEAM turbines as manufactured today are entirely reliable machines, and under ordinary conditions and with reasonable care give little trouble. Like any other piece of machinery, human or otherwise, small turbines are not infallible and there are times when they show symptoms of trouble. If these symptoms are diagnosed correctly when they first appear, and if steps are taken immediately to correct the cause of the trouble, no serious difficulties will ensue; but if the first slight indications of trouble are disregarded, trouble of a more serious nature may develop which will finally put the turbine out of service.

One of the most common indications of trouble in a turbine, is undue vibration. Turbine rotors are delicately balanced, and the slightest amount of wear, distortion, or breakage is sufficient to throw the machine out of balance.

In Fig. 1 is seen a rotor of a 150-kw. generator unit, three of which came back to the shop for repairs. The face surface of the bucket wheel was badly worn as if it had been constantly rubbing against the nozzle and several blades were broken out at A and A′. The blade edges were all worn with deep V-shaped cuts as seen at B.

In Fig. 2, the shaft is badly scored on both bearings at e-e. In this type of machine, the edges of the blades in the wheel, when new, are a little thick on one edge as shown by the shaded portion, c, and the other edge is beveled over a little sharper and thinner as at d, the direction of flow of the steam impinging against the end of this thick edge. It is supposed that superheated steam, which was used, impinging against this edge in the center of the blade at X, cut the V-shaped grooves and that vibration caused the wear on the face of the rotor and face of the nozzles, shown in Fig. 3. Two of these nozzles had the partitions broken and bent as at d and e. It is supposed that a slug of water passed through these two nozzles at high velocity, which broke the partitions out and also broke the blades in the rotor. This could happen if the steam separator on the steam line became flooded from accumulated condensate due to a plugged up drain pipe or inactive trap or a priming boiler.

In this instance, a new rotor and shaft with new nozzles were made and the turbine assembled and tested on the test plate. Figure 4 shows the turbine and generator assembled on base plate with connections from the governor to throttling and emergency stop valve omitted.

There are many causes of vibration in the turbine; in fact, almost any minor trouble, if not corrected, will develop vibration which will grow to such serious proportions as to cause the rotor to rub the nozzles. There is no machine more neglected than the one which does not require constant attention and this is why so many small turbines have to be returned to the manufacturer to be repaired. Most such machines are placed in the care of an attendant, whose knowledge of engineering consists mostly in starting, operating and stopping such units, and not knowing what the consequences may be, he pays little attention to slight indications of vibration. This vibration usually grows until the machine becomes so badly worn in all parts that it is cheaper to send it to the shop for repairs than to repair it at the plant. It generally requires a skilled engineer or mechanic accustomed to turbine operation to detect the causes of vibration from the symptoms indicated and to adjust it for its rated capacity after the repairs have been made.

Some turbines have oil wells in the bottom of the bearings with chains or rings to carry the oil up over the shaft. The generator bearing will generally be supplied with an oil pump oiling system. In Fig. 4 the oil pump may be noted on the end of the shaft at d. The suction and discharge passages cast in the side of the bearing stand is shown at T.

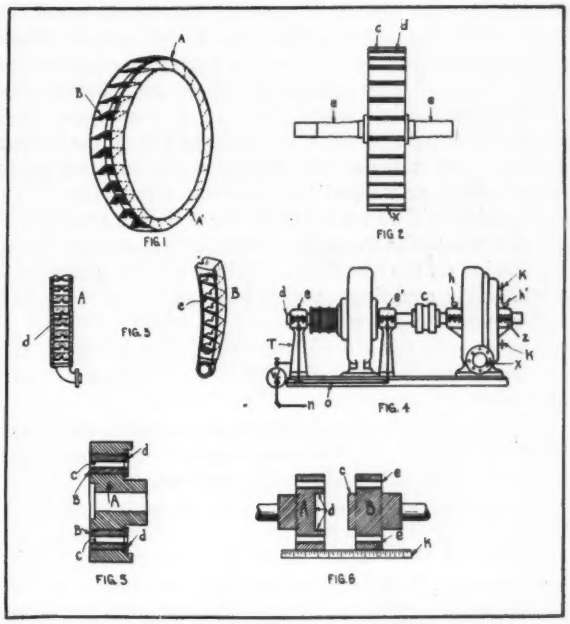


FIG. 1. SKETCH OF A DAMAGED TURBINE ROTOR
FIG. 2. FACE VIEW OF ROTOR WITH SCORED SHAFT
FIG. 3. FACE AND EDGE VIEW OF NOZZLE
FIG. 4. TURBINE AND GENERATOR, DIRECT CONNECTED
FIG. 5. ONE-HALF OF COUPLING WITH INSULATORS IN POSITION
FIG. 6. METHOD OF CHECKING ALINEMENT OF COUPLING

OIL TROUBLES

The bottom portion of the generator bearing stands are hollow and used as oil reservoirs connected together by a pipe allowing the oil to be used over and over, the oil being pumped from the reservoirs through the cooling tank W, then to the generator bearings e-e, through the pipe O, and the return runs down to the reservoirs. At N is shown the cold water supply pipe to the cooling tank. The oil has to withstand a very high temperature and will therefore thicken up if allowed to get cold. As a consequence, when the turbine is not running the

pump cannot draw it or pump it around through the pipe lines to the bearings. For this reason the cooling tank is used mostly in hot weather. The oil in these bearing bases should be examined at regular intervals to see that it does not become filled with dirt or other impurities. These impurities often get into the oil when adding new oil packings at the ends of the bearings. On the turbine bearings there is a thermometer at z-z at all times to indicate the temperature of the bearings which will generally rise considerably as the generator warms up; but when any extreme heating occurs, it is immediately detected by the thermometers, before it reaches the danger point.

Often these thermometers are allowed to get out of order and become broken. The oil gage glass becomes clogged up and shows oil apparently at the right height, while the well under the bearing is empty or so near empty that the bearing, becoming hot, wears away the babbit, thus causing vibration which will have a serious effect on the coupling between the turbine and generator shaft. Even if the bearings do not get hot, the machine running at a high speed, will in a few years wear the babbit down and allow the rotor to rub on the bottom nozzles. An unbalanced armature or turbine rotor, sprung shaft, or warped turbine rotor often cause vibrations which will cause sufficient wear in the bearings to put the two shafts out of line, bringing constantly changing strains on the couplings. This is more common with semi-flexible couplings, which have to be used on electric generators direct connected to turbine shafts. If reduction gears are used, this wear of bearing babbits and couplings will cause trouble due to gears meshing improperly.

In Fig. 5, A is a half portion of a semi-flexible insulated coupling. B, B are semi-hard rubber insulating bushings with a brass thimble bushing. C, C in the center bolt holes, d, d are hard fiber or rubber insulating washers, all of which will wear with vibration or misalinement of the shafts. Some direct connected turbine units have only three bearings and vibration, if not immediately corrected, will multiply so rapidly, when once it commences, as to put the turbine out of service in a short time. Such units require a rigid coupling, usually of the plug and socket type, and must also be insulated.

Doing Repair Work in the Plant

In most steam plants there are no proper tools to make repairs on a turbine unit, also, the attendants, unless skilled engineers, have not sufficient experience as mechanics to do the work. Furthermore, since turbine operation requires little or no attention, the operator is loaded up with numerous other duties which do not allow time to overhaul the turbine when necessity demands it. If he does take time to make the repairs, the lack of proper tools often causes more trouble which requires the machine to be sent back to the manufacturer to be put in working order.

One type of repair requiring knowledge and tools is the disconnecting of the rigid coupling. When clamps and draw bolts are not at hand an attempt is usually made to separate the two flanges of the coupling with a very blunt cold chisel. Even if this proves successful, it will batter the face edges of the couplings so that when again connected they will not come together in perfect alinement. The battered spots then have to be carefully removed from the face of the coupling and the plug or socket portion, if bruised in handling, must be smoothed up also.

In removing such bruises, there is some danger in removing too much of the metal from the face of the coupling, which will, when coupled together, put it out of alinement in the opposite direction and make it necessary to true up the coupling in a lathe.

Even new machines set up on the test plate, inspected and started in operation, will develop vibration which, on careful examination in checking up alinement, shows the face of the flange to be out of perfect right angle with the shaft by 0.001 in. if the other half portion is out an equal amount and coupled together on the same side, the ends of the shafts will be out considerably. The shafts must either spring at the coupling or vibrate at the ends. In checking up the coupling for alinement, a straightedge scale is placed lengthwise across the edges of both coupling flanges as at K in Fig. 6. A is the socket portion and d shows the insulator in the socket. B is the plug portion with c the plug and e, e are the semi-hard rubber bushings in the bolt holes. When the turbine is connected to blowers or other apparatus not generating electric currents, the insulators are not needed in the couplings.

To check the alinement of the shaft and bearings, the face of the two parts are left apart enough to insert a feeler gage about 0.0015 in. thick and pass all the way around the coupling. If equal all the way around, it is as close to alinement as practical and will run. The bolts can be drawn up tight. In many cases at the first sign of vibration the coupling should be disconnected and examined, as it will indicate where the misalinement is, with the least labor and time. There are other methods, however, of checking up the alinement, which are used by engineers who are not thoroughly familiar with use of micrometers.

Sprung Shaft or Rotor

If a shaft is sprung or the rotor warped, the best way to find out the amount it is sprung is by suspending it between the centers of a lathe if one is available. If not, centers can be attached to the bearings with the rotor raised up from the case and the shaft and rotor revolved. With a pencil set or held rigid at several points along the shaft, the high parts will be marked. If the shaft is not badly sprung, it may then be hammered with sledge and wood or copper block to prevent bruising, or, better still, if there is a screw or hydraulic press convenient put it under the press and straighten it. Put it back in the lathe centers and try a straightedge along its length, all the way around, or try the pencil test again. If straight, the pencil will make a mark all around the shaft. The same pencil test can be made with the rotor by holding it against the edge of the rim. When the rotor is revolved the pencil will mark the high spots. This method does not caliper the amount the shaft or rotor is sprung, and an indicating micrometer is more accurate, which when held rigid against the circumference of the shaft or edge of the rotor will indicate plus in thousands of an inch on the high spots and minus on the low spot. This indicates accurately the amount the shaft or rotor has to be sprung to get it back straight.

In Fig. 1, the rotor has the bucket blades on the face circumference; but many rotors have the blades on the side of the rotor which makes them more subject to

warping from the effects of a slug of water or when a blade gets loose and jams between the rotor and the nozzle. The clearance space between the rotor and nozzles in some turbines is only a few thousandths of an inch. When thrust bearings are used, there is often sufficient wear in them to allow the blades to rub. This is indicated by a rasping sound in the turbine case when starting the turbine, but is not so noticeable when the turbine comes up to speed. The thrust bearing must be properly adjusted to prevent end play of the shaft.

If this bearing should become sufficiently overheated to cause the rings to become scored and cut, it will have a period of vibration of its own which will cause an oscillating motion of the shaft, causing the rotor to rub at various points. The packing will also wear faster with this end thrust. If the bearings have worn down to any considerable degree, the packing will also be worn an equal amount and the oil seals on the ends of the bearings will leak and allow the oil to be thrown out. These troubles can be cured by the engineer by putting in shims under the babbit boxes or machining the glands and adding new packings. The adjustment of the thrust bearings, however, must not be changed unless one has sufficient experience to be sure of the measurements, as a mistake here will damage the rotor by rubbing.

In shimming up the bearings, care must also be used to see that the range of movement of the governor levers is not changed to such an extent as to cause an increase of speed above the proper rate. All joints of levers and pins connected to the governor weight and valves must be kept clean from burnt grease and dirt and not allowed to become warm to any great degree of looseness or the governor cannot operate properly.

Unbalancing can often be found in the generator armature, the ends of the coils becoming bent to one side and the accumulation of dirt adding more weight to that set of coils. Engineers are in a position to detect unbalancing, which occurs while the machine is in operation, and can thereby determine and prevent this to some extent by keeping the armature coil clean. In Fig. 4, p and p′ are bolts which keep the babbited shells on the shaft in position. As the babbit becomes worn, it is customary to remove a shim from between these shells to prevent the shaft from jumping or vibrating. This causes trouble to appear in the generator, either as vibration or apparent unbalancing and in a unit of only three bearings it will cause the shaft to rock over the middle bearing. In all cases when shims are removed from one bearing, they must be removed also from the other bearings to keep the shaft level and in line. This work can be done by an engineer in the plant if he has had machinist experience in using micrometers and in alining and leveling shafts. There are so many men who have not this experience that manufacturers have to be consulted and often the machines are sent to the shop. In the case of the machine shown in Fig. 4, the armature exploded. It was rewound, reassembled and found to be out of balance; it had to be rewound three times before it was properly balanced. Hence it is better to send the machine to the shop and save time and freight charges. Even experienced armature winders have trouble in keeping the winding balanced and one who can do such work is considered a skilled expert in his line. Engineers can, however, scrape and fit the babbit bearings, and replace packings and should have sufficient experience to detect minor defects before starting the machine, which would save the employer much delay and expense for repairs.

Since the coupling is an indicator by which it is most convenient to determine whether the cause of vibration is in the turbine or in the generator it is important to have it in perfect condition. When vibration occurs, the coupling is always the first thing to examine. By disconnecting the coupling and starting the turbine rotor, it is easier to detect vibration caused by a warped rotor wheel than with the generator connected. The sound of rubbing will be more distinct and if wear of the shaft shoulders against the ends of the bearing babbits has been severe, there will be some room for end play of the shaft. The warped rotor will cause increased oscillation so rapidly as to appear in the form of vibration very hard to detect when the shaft is coupled together.

When weights have been added to the armature or turbine rotor for the purpose of balancing them, they should be perfectly secure and no chance of working loose. A weight riveted to a rotor solid today may be loose tomorrow. Such weights move with inertia as the load changes on the machines and when one goes adrift, it takes the rotor and nozzles with it.

Proportioning Electric Power to Exhaust Steam Required

MARKET FOR SURPLUS ELECTRIC POWER ENABLES EXHAUST STEAM TO BE PRODUCED AT LOWEST POSSIBLE COST. BY HENRY R. GOULD

RECENTLY a striking example of inefficiency in operation of station auxiliaries, arising from changing load conditions, came to the writer's attention. While the conditions surrounding this particular plant were peculiar, the lessons therefrom may be of general application and interest.

A 10,000-kw. steam turbo-generator station was furnishing electricity and steam for manufacturing to a large factory with a load of about 5000 kw. at that time. Its boilers were 600-hp., water-tube, oil fired. Two main turbines were operated non-condensing, and one extracting, at 14 lb. back pressure. The auxiliaries, pumps for oil and feed water, fans, etc., were all designed to be driven by non-condensing impulse turbines, operating at the same back pressure. There was a credit to the power department for all low-pressure steam, of 90 per cent of the cost of live steam, and the demand was so large that considerable live steam had to be expanded through reducing valves to keep up the pressure.

To remedy this uneconomical condition, a market was found for such excess power as would normally be produced by the three turbines, operating well up to capacity. At that point, several thousand kilowatts could be sold, and the steam demand supplied without the normal use of live steam. Steam balance was a vital problem. There must be a minimum use of high-pressure steam for manufacturing, the minimum amount of steam condensed, and no open exhaust.

It was then realized that the power department was getting poor returns from the undoubtedly large amounts of steam used by the auxiliary turbines. Inves-

tigations and tests were therefore begun with surprising results, as tabulated:

Unit	Rated hp.	Rated steam per hp., lb.	Actual hp.	Actual steam per hp., lb.	Actual steam per hour, lb.
Large pump...	65	70	20	105	2100
Small pump ..	15	120	4	190	760
Fan	52	77	65	77	5000

Now, as the non-condensing turbines had a water rate of 35 lb. per kw.-hr. and the extraction turbine a rate of 24 lb. per kw.-hr., it was evident that the same steam through these more efficient turbines would produce a considerable amount of power for sale.

For instance, the 2100 lb. of steam per hr. put through the large pump turbine produced 2100 lb. of exhaust steam and 20 hp. to drive the pump. Putting this steam through the other turbines the savings could be computed as follows, assuming that power could be sold at $0.010 per kw.-hr.:

(1) Steam to non-condensing turbines, per hr...2100 lb.
Returns from these, per hr.
18 kw.-hr. for the switchboard to produce 20 hp. motor output.
42 kw.-hr. for sale.
2100 lb. exhaust steam.
Annual value of power available for sale $42 \times 60 \times 52 \times 0.010$ $1310
Estimated cost of equipping with d.c. motor and variable speed control.....$1600
Annual return on investment..... 82%

(2) If load conditions at times permitted utilizing this steam through the larger and still more economical extraction turbine, an even larger saving resulted. To produce from this unit the 2100 lb. of exhaust steam per hour necessary to maintain the steam balance, it was necessary to generate an extra 1100 lb. per hr. of high pressure steam, which was put through the turbine and condensed. The fuel cost of generating this extra steam was therefore an extra charge against the gross gain or saving made. The figures are:
Steam to extraction turbine, per hr.....3200 lb.
Returns from turbine, per hr.:
18 kw.-hr. at the switchboard to produce 20 hp. motor output.
115 kw.-hr. for sale.
2100 lb. exhaust steam.
Annual value of power available for sale $115 \times 60 \times 52 \times 0.010$.....$3588
Less annual cost of fuel to generate the additional steam $1100 \times 60 \times 52 \times 0.34$.... 1167

Net return from sale of power..... $2421
Annual return on investment.....151%

Savings on the other units were of similar proportions and the combined effect of all the savings was sufficient to indicate a reduction in the station's net cost per kw.-hr. of over 2 per cent. Of course, this plant was in a somewhat exceptional position, and the percentage returns indicated probably would not obtain in all cases. The pump turbines were considerably overpowered, the fan turbine was of slow speed; all were uneconomical. The whole proposition, likewise, has its limitations, as to how many and what auxiliaries it may be wise to motorize.

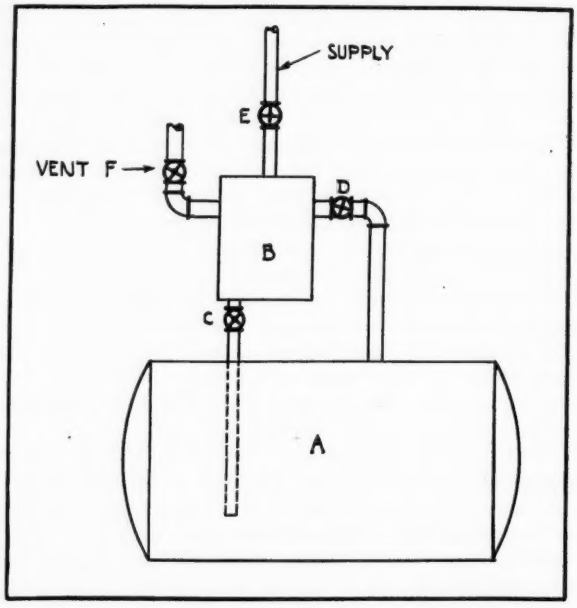
A Simple Boiler Feeder*

By W. F. Schaphorst

IN THE accompanying illustration is shown a method for feeding water into a boiler under pressure without the use of an injector or pump of any kind.

A is the boiler that is to be fed with water. B is a tank or vessel capable of withstanding as high steam pressure as the boiler itself.

Close valves C and D and open valve E and vent F so that tank B will fill with water through the supply pipe. Then close valves E and F and open valves C and D. Steam will enter B through the valve D and the water from B will flow downward through C due to gravity because the vessel B is now nothing more or less than a part of A and since water naturally seeks its level in pressure vessels as well as in the open atmosphere it will flow downward, B will empty and the water level in A will be raised. After B is empty, close C and D again and open E and F and repeat the foregoing process. The writer is informed that a boiler feeding device of this kind is now on the market and that it is proving to be a success on large as well as small boilers. Its simplicity should certainly attract attention.



A SIMPLE DEVICE FOR FEEDING WATER INTO A BOILER

It is well to note in this connection, however, that some steam is required to do this work. When all of the water is emptied out of B, that vessel is full of steam, hence the volume of steam used is equal to the volume of water fed into the boiler. It is therefore a simple matter to compute the amount of steam required to feed a given quantity of water into a boiler with this apparatus.

DON'T EXPECT a boiler to run efficiently if the tubes are full of scale on the inside and covered with soot on the outside.

*All rights reserved by author.

Lowering Oxygen in Water Reduces Corrosion

Laws Governing Amount of Gas Water Will Dissolve and Means of Deaerating Water. By Charles E. Colborn

CORROSION of the internal surfaces of boilers and economizers has always been an objectionable incident to the generation of steam. The removal of the scale-forming elements in the feed water has in many cases resulted in an increased amount of corrosion. It is generally recognized that the remedy for this corrosion is to reduce the amount of oxygen in the feed water. Several theories have been advanced to explain the action of the dissolved oxygen in corroding the metal of boiler and pipe line surfaces. Although there is no general agreement as to which of these theories is right, all are fairly well agreed that the oxygen is the active agent in this corrosion. In one test it was noticed with steel tube economizers that, with an initial content of about 4 c.c. per liter, a reduction of 75 per cent in the oxygen content was effected during the passage through the economizer. In the case of cast-iron economizers, similar reductions but of a lesser magnitude were found.

Gases such as oxygen and nitrogen obey a single law with regard to the amount that will be dissolved in any given solvent. This law is known as Henry's law and states that a given quantity of the liquid will dissolve at constant temperatures quantities by weight of gas which are proportional to the pressures of the gas. The amount that is dissolved with a given gas pressure varies with the temperature of the liquid. This means that the amount of oxygen or air that can be dissolved in the feed water in any case is determined by the temperature of the water and the air or oxygen pressure. The amount of oxygen dissolved can be reduced by reducing the partial air pressure. According to Dalton's law, the partial pressure of air in any closed vessel containing air and water will be the total pressure minus the water vapor tension. In mechanical deaerators, the air is removed from the water by either heating the water until the vapor tension becomes almost as great as the total pressure or by reducing the total pressure to a point nearly as low as the vapor tension. In either case, the difference between the vapor tension and the total pressure is determined by the density of the air which is kept extremely low by venting.

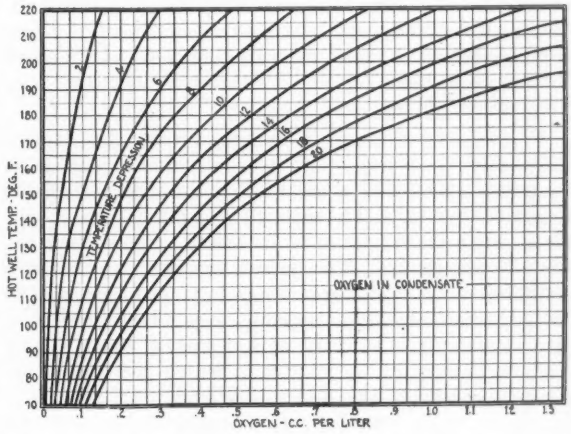
In cases where the air pressure is reduced over a large body of water, there is a delay in the giving off of the oxygen. In deaerators, the removal of the oxygen is accelerated by some mechanical action which tends to disengage the bubbles of gas. In one type of deaerator, this mechanical action is secured by spraying the water into a chamber in which the total pressure is lower than the pressure corresponding to the temperature of the entering water. The agitation resulting from the flashing of the vapor from the water causes the water to give up the dissolved gases rapidly.

Keeping Oxygen Out of Feed Water

Several form of apparatus have been developed for eliminating or deactivating the oxygen in boiler feed water. This apparatus in general has given good service, but is expensive and in many cases complicated. The feed water can in most cases be kept practically oxygen free without the use of special apparatus for the removal of oxygen, if the equipment is kept in such condition that the oxygen cannot enter the feed water system. Oxygen can enter a feed water system at any of the following places:

a. By being dissolved from the air in the surface condensers.

b. By the entrainment of bubbles of air in the condensate which is being removed from the condenser. This air is dissolved by the water when the pressure is



VARIATION OF OXYGEN IN CONDENSATE WITH CHANGES IN HOT WELL TEMPERATURE AND TEMPERATURE DEPRESSION

increased as the water passes through the condensate pump.

c. By the leakage of air into the water through the glands of the condensate pumps.

d. By the water being exposed to the air in the surge tank, in heaters or in V-notch meters.

e. By the addition of make-up water containing large quantities of oxygen.

The amount of oxygen that can be dissolved by the water in the hot well of a condenser is proportional to the oxygen pressure in the mixture of water-saturated air above the water in the hot well and through which the condensate drips. The oxygen pressure in this part of the condenser is determined by the amount the temperature of the condensate water is depressed below the temperature of the steam entering the condenser. The illustration shows the quantities of oxygen required to saturate the water, with various condensate temperatures and amounts of depression of the condensate temperature.

In modern condensers, in which the depression of the condensate temperature is small, the amount of oxygen in the hot well water will also be small. It has been frequently stated in technical literature that large amounts of oxygen in the condensate are due to excessive air leakage into the condenser. The amount of oxygen in the condensate is by no means an index of the amount of condenser leakage. The curves show that with reasonably good vacuums the oxygen that can

be dissolved by the water in the condenser hot well will not be great even with a large depression of the condensate temperature. In tests made upon a large surface condenser, it was found that the amount of oxygen in the condensate was reduced when sufficient air had been admitted to the condenser to reduce the vacuum approximately 2 in. of mecury.

Throttling Pump Discharge Reduces Oxygen in Water

The amount of oxygen that is entrained in the form of bubbles in the water entering the condensate pump can be determined by comparing the oxygen found in the condensate when the pump discharge valve is throttled so that the water level rises in the hot well, with the amount of oxygen found with the pump unthrottled. The amount of oxygen entering the condensate by this air pump action of the condensate pump is usually small and can be reduced by slightly throttling the discharge valve of the pump.

Of the oxygen found in the water from the condensate pumps, the greater part is usually due to leakage at the pump glands. The air leaking through the glands is carried by the water through the pump and is dissolved when the pressure is increased at the discharge side of the pump. The gland leakage can be kept negligibly small if the glands are kept properly packed and adjusted. The glands should never be tightened so tight as entirely to stop the flow of sealing water. There are now pumps on the market in which the glands are under pressure of the pump discharge. In pumps of this type, there would be no danger of affecting the oxygen content of the water in its passage through the pump.

Aeration of the water in the heaters can be prevented by closing the heater so as to make it airtight. If possible, sufficient steam should be added to maintain the water temperature at 212 deg. or above and a slight amount of steam should be vented from the heater. When it is not possible to maintain the feed water as high as 212 deg., greater pains must be taken in making the heater airtight and an ejector should be installed to remove any air that might leak in.

Aeration of the water in the surge tank can be prevented by closing the top of the tank and admitting exhaust steam to the space above the water.

In plants where evaporator condenser units are used to distill the makeup water, the oxygen is almost completely removed in the evaporator condenser. Where this equipment is not used, the makeup water can be deaerated by spraying it into one of the main turbine condensers. The water should be sprayed in below the tube bank if possible so as not to foul the outside of the condenser tubes. If the makeup water is at a temperature higher than that corresponding to the pressure in the condenser, as it would be if it were first treated in hot-process water softeners, the deaeration will be nearly complete.

It has been demonstrated that the corrosion of piping and economizers is proportional to the amount of free oxygen in solution. Feed water to steel economizers should contain less than 0.2 c.c. of oxygen per liter. No trouble should be experienced in maintaining the oxygen content this low in condensing plants having closed feed water systems. The following results are obtained in a large power plant in which the makeup water is distilled in evaporator condenser units:

Av. oxygen in main turbine condensate	0.05 c.c. per liter
Av. oxygen in condensate from evaporator condensers	0.20 c.c. per liter
Oxygen in feed water leaving heater	0.08 c.c. per liter
Per cent makeup	4.00
Per cent condenser leakage	0.5

Chart for the Design of Gas Passages

Explanation of Chart for Reading Areas of Gas Passages in Boilers and Furnaces. By M. E. Yeager*

OPERATION of modern power plants at high capacities has brought to the attention of engineers the necessity of having flue gas passages generously designed. The accompanying chart is an extension and refinement of engineering information on this subject. The operation at higher boiler capacities has resulted in increased flue gas temperatures and velocity ranges. This chart facilitates determining the construction requirements within this range of high temperature and velocity.

Variable factors which must either be known or assumed in the use of this chart are: boiler horsepower developed, pounds of coal per boiler horsepower hour, B.t.u. value of the coal per pound as fired, total air used per pound of coal in terms of per cent excess over that theoretically required and temperature and velocity of gases at the point considered.

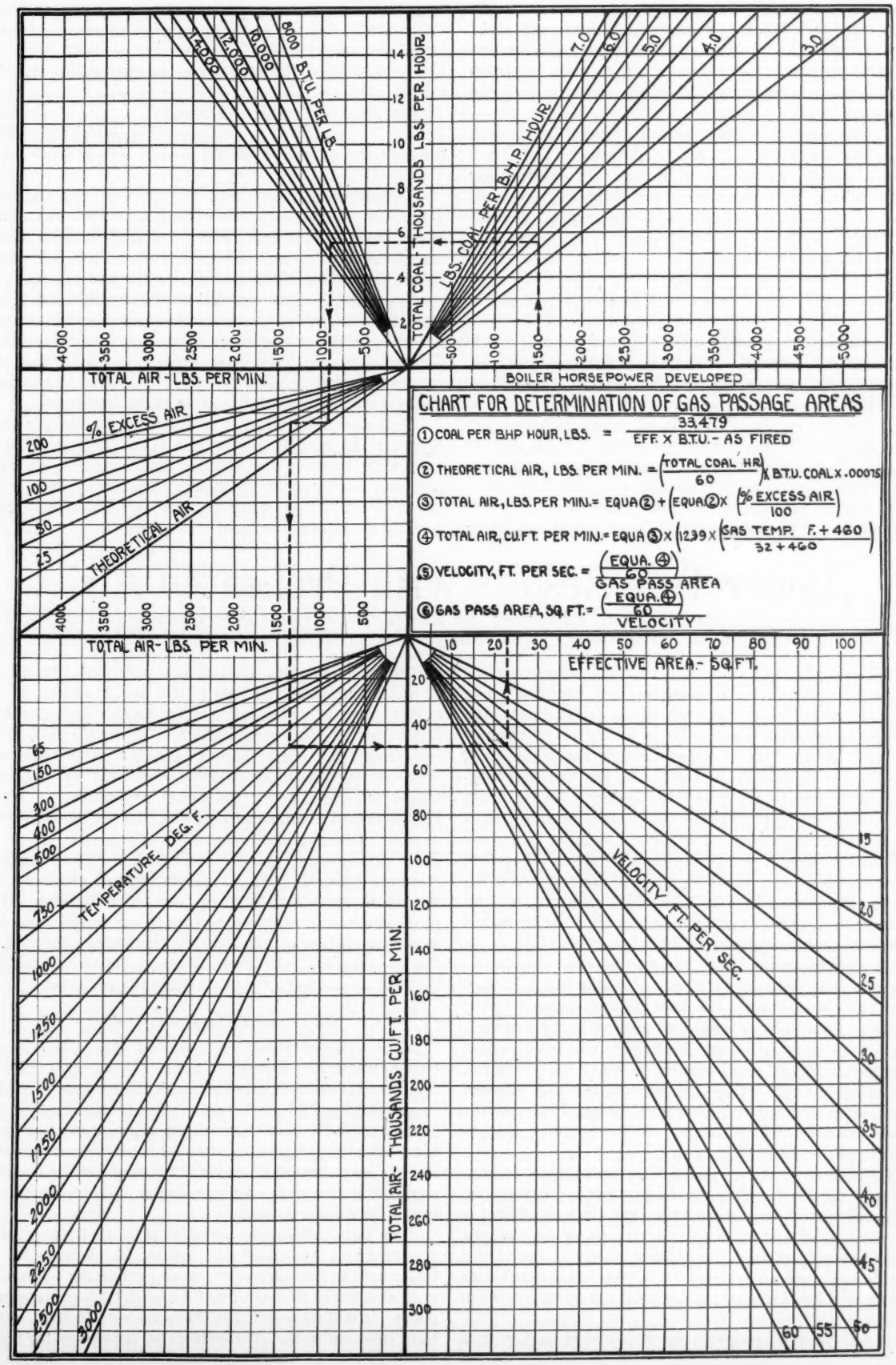
Knowing or assuming the combined boiler and furnace efficiency and B.t.u. value of the coal per pound as fired, the coal required in terms of pounds per boiler horsepower hour = 33,479 ÷ (efficiency × B.t.u. value coal per pound as fired).

With the pounds of coal per boiler horsepower hour known or determined as explained and the other factors known or assumed, the use of the chart may best be illustrated by example.

Assume 1500 b.hp., coal of 13,000 B.t.u. heating value per pound as fired, 70 per cent combined boiler and furnace efficiency, 3.7 lb. coal per b.hp.-hr. determined from the equation for these conditions, 50 per cent excess air supply over theoretical requirements, 1000 deg. F. gas temperature at point to be considered and 35 ft. per sec. gas velocity.

On the chart, locate 1500 hp. on horizontal scale in upper right hand block. Project vertically to a point between the oblique coal lines representing 3.7 lb. per b.hp.-hr., continue horizontally to the left to the 13,000 B.t.u. diagonal line and down vertically to the theoretical air line. Proceed to the left horizontally to the oblique line representing 50 per cent excess air and down

*Stoker Engineer, Pittsburgh District, Westinghouse Electric & Manufacturing Co.


14,000
12,000
10,000
8000 B.T.U. PER LB.
TOTAL COAL - THOUSANDS LBS. PER HOUR
7.0
6.0
5.0
4.0
3.0
LBS. COAL PER B.H.P. HOUR
BOILER HORSEPOWER DEVELOPED
TOTAL AIR - LBS. PER MIN.
% EXCESS AIR
200
100
50
25
THEORETICAL AIR
CHART FOR DETERMINATION OF GAS PASSAGE AREAS
① COAL PER B.H.P. HOUR, LBS. = 33,479 / EFF. × B.T.U. - AS FIRED
② THEORETICAL AIR, LBS. PER MIN. = (TOTAL COAL PER HR / 60) × B.T.U. COAL × .00075
③ TOTAL AIR, LBS. PER MIN. = EQUA ② + (EQUA ② × (% EXCESS AIR / 100))
④ TOTAL AIR, CU. FT. PER MIN. = EQUA ③ × (12.39 × (GAS TEMP. F. + 460) / (32 + 460))
⑤ VELOCITY, FT. PER SEC. = (EQUA. ④ / 60) / GAS PASS AREA
⑥ GAS PASS AREA, SQ. FT. = (EQUA. ④ / 60) / VELOCITY
TOTAL AIR - LBS. PER MIN.
EFFECTIVE AREA - SQ. FT.
65
150
300
400
500
750
1000
1250
1500
1750
2000
2250
2500
3000
TEMPERATURE DEG. F.
TOTAL AIR - THOUSANDS CU. FT. PER MIN.
VELOCITY FT. PER SEC.
15
20
25
30
35
40
45
50
55
60

vertically to the 1000 deg. F. temperature line in the lower left-hand block. From this point read right horizontally to the 35 ft. per sec. diagonal velocity line and then up vertically where 23 sq. ft. will be determined as the approximate effective gas pass area at the point considered.

Formulas given in the intermediate right-hand block on the chart are not required in the use of the chart, with the possible exception of the first for determining pounds of coal per boiler horsepower hour, but have been added as explanatory and for ready reference. A brief explanation of these formulas follows:

(1) Lb. coal per b.hp.-hr. $= \frac{33{,}479}{\text{Eff.} \times \text{B.t.u. coal (as fired)}}$

This formula has already been referred to and needs no further explanation. It may be of interest, however, to explain the derivation of 33,479 which is the B.t.u. per b.hp.-hr. and is obtained by multiplying 34.5×970.4, the former being the lb. of water required per boiler horsepower hour when evaporated from and at 212 deg. F. and the latter is the latent heat of evaporation or B.t.u. required to evaporate one pound of water at 212 deg. F. to steam at the same temperature.

The volume of a given weight of any gas increases or decreases directly as the absolute temperature (deg. F. + 460). At 32 deg. F. and atmospheric pressure (14.7 lb. per sq. in.) the volume of one pound of air is 12.39 cu. ft. From these facts, the derivation of equation (4) will be readily seen.

(2) Total air, cu. ft. per min. $=$ total air, lb. per min. $\times 12.39\left(\frac{\text{Gas Temp. deg. F.} + 460}{32 + 460}\right)$

Equations (3), (5) and (6) are self-explanatory. Equations (5) and (6) have been added to show how either the gas pass area or velocity may be determined mathematically, when either is known, together with the total air flow in thousands of cubic feet per minute.

Results which may be obtained from this chart are all within the limits of error permissible in boiler room practice. Great care must be used, however, in assuming the variables as it is only reasonable to expect that the results obtained from the chart will be only as accurate as the data upon which the calculations are based.

It is extremely difficult to approximate the gas temperatures at various points in a boiler setting, as they vary with furnace temperature, boiler rating, air supply, and condition of the heating surfaces. It is well, therefore, to assume temperatures as high as may reasonably be expected to exist at any time.

Boiler Test Results with Preheated Air*

MARKED INCREASE IN OVERALL EFFICIENCY AND BETTER COMBUSTION CONDITIONS RESULT FROM USE OF PREHEATED AIR IN COLFAX STATION OF DUQUESNE LIGHT CO. BY C. W. E. CLARKE†

IT WILL be noted in Fig. 1 that the installation of the preheaters, ducts, dampers and fan was made without any addition whatever to the boiler house, the changes required being simply an alteration of the uptake and the cutting of holes through the boiler-room floor for the preheated air duct. As the drawing indicates, the connection between the common forced draft duct normally supplying the boilers has been retained and is cut off by means of a shut-off damper when the preheater system is in operation. The air is taken into the preheater from the boiler room directly over the boiler, as indicated in Fig. 1, carried down the duct by the fan and discharged into the stoker wind box by means of two ducts extending on either side of the boiler. A 7½-ft. double-suction, plate fan was used on this installation so that pressure of 11 in. of water could be carried if found necessary, no exact data on pressure drop through the preheater and duct system being available at the time of installation. Belt drive was used so that various fan speeds could readily be obtained. This fan is now being replaced by a more compact and efficient direct-connected motor-driven unit designed to meet the exact conditions which obtain. The duct work is insulated to prevent loss of heat.

In Fig. 2 is shown the preheater which was constructed by the Combustion Engineering Corporation, following in general the designs of the Underfeed Stoker Co., Ltd., of London, England, excepting in the one important respect that the heating elements are placed back to back and separated by a bypass space fitted with a damper. The preheater is designed for operation when the boiler is evaporating the equivalent of 144,000 lb. of water per hour from and at 212 deg. F., which is the normal rate at which the Colfax boilers are operated. It was not considered worth while to install preheating equipment for ratings higher than this as extreme ratings are emergency conditions and the savings obtained would not justify the additional expenditure. When high overload ratings are required, the preheater bypass damper is opened to allow more uptake area.

It will be noted that the arrangement of two banks of elements back to back, as shown, makes available the maximum surface in a given vertical dimension. The preheater was guaranteed to supply air to the wind box at a temperature of 235 deg. F., the boiler operating at the evaporation rate given above. The tests indicate that the equivalent of this was actually obtained.

AIR LEAKS THROUGH PREHEATER

Although the preheater is riveted and calked, it was found impossible to make it tight. There is therefore a considerable infiltration of air into the flue gas resulting in an exaggerated reduction in temperature after the preheater and a reduction in actual preheater performance. A new design using welded construction throughout has been developed and will be used for the installation now under way. An increase in preheater efficiency should be gained thereby.

The tests, which are the subject of this paper, were carried on to determine the increase in efficiency of the boiler unit in question when using preheating equipment over and above that obtained without preheating equipment. It will be noted that there are certain local conditions inherent in the arrangement of apparatus for boiler No. 9 at Colfax which may not obtain for a similar in-

*Abstract of paper presented at annual meeting of the A. S. M. E. Dec. 3 to 6.

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stallation in another place. One point we have in mind in this connection is that the temperature of the inlet air to the preheater is somewhat high due to the fact that the preheater suction is located directly above the boiler tested. It is our desire to present actual results rather than any calculations which would bring these conditions to a common basis.

These tests were run on what is known as boiler No. 9 at the Colfax station during the period from July 14 to Aug. 4. The runs were each of 24 hr. duration and each series, one series with and one without the preheater, was continuous for the periods of 8 to 13 days respectively. Better results doubtless could have been secured had a definite effort been made to establish high efficiencies. No such attempt was made. Throughout the test the boiler and stoker equipment were handled by the regular boiler-room force, with some assistance from the Westinghouse stoker engineers. The operation did not differ from the regular daily method, with the exception of the manipulation of the wind box dampers to give as nearly a constant rating as possible during the period of each test. In short, every effort was made to have the test results representative of day to day operation.

The tests were directed and supervised by Dwight P. Robinson & Co., Inc., assisted by representatives of the Duquesne Light Co., The Babcock & Wilcox Co., and the Westinghouse Co. and were conducted in so far as possible in accordance with the A. S. M. E. test code.

The boiler and preheater equipment is best described by the following tabulation:

Boiler No. 9 Colfax Station

Babcock & Wilcox cross-drum, water-tube type boiler.
Installation, 1922.
22,914 sq. ft. heating surface.
275 lb. per square inch gage steam working pressure.
Normal rating (100 per cent) 79,053 lb. of water from and at 212 deg. F.
20 tubes high, 51 tubes wide, 4 in. diameter, 20 ft. long.
60 in. diameter drum, 34 ft. 1 in. long.
Setting 30 ft. 1 in. wide inside of walls.
Babcock & Wilcox superheater containing 2999 sq. ft. heating surface located in first pass of boiler.
Furnace volume 7500 cu. ft.
Westinghouse Company underfeed stoker, 17-retort, 20-tuyere, extra long stoker with side-wall tuyeres.
Clinker grinder, double-roll, separate drive on each side of boiler.
29 ft. 10¼ in. wide by 13 ft. 6 in. effective length of grate.
218.31 sq. ft. projected area of grate.
163.89 sq. ft. grinder pit section.
11,200 sq. ft. preheating surface.

Comments on Test Data and Results

Gas temperatures above the preheater, shown in Fig. 5, with the preheater in operation, are unduly low because of air infiltration through the preheater.

It will be noted that the flue gas temperature before the damper, Fig. 5, is approximately 15 deg. lower on the average than the same reading in the data without air preheater. This is due to the higher furnace temperature and better CO_2, resulting in a more efficient heat transfer in the boiler.

The temperature of the preheated air, Fig. 4, was taken at the fan discharge rather than the preheater discharge because the fan decreased the possible error due to stratification. As the preheated air duct between the preheater and the fan suction is effectively insulated, the drop in temperature between these two points was considered negligible.

The variation in gas composition across a section of one of the passes varies as much as 2 per cent in actual value of CO_2 and from the front to back of the pass as much as 4 per cent in actual value of CO_2. With an average CO_2 value of 12 per cent these figures correspond to a variation in excess air of 22 per cent and 50 per cent respectively, so that it is extremely difficult if not impossible to arrive at results actually representative of average conditions across the pass. The analyses of dry gases before the damper show a complete absence of carbon monoxide, hydrogen and hydro-carbons. The

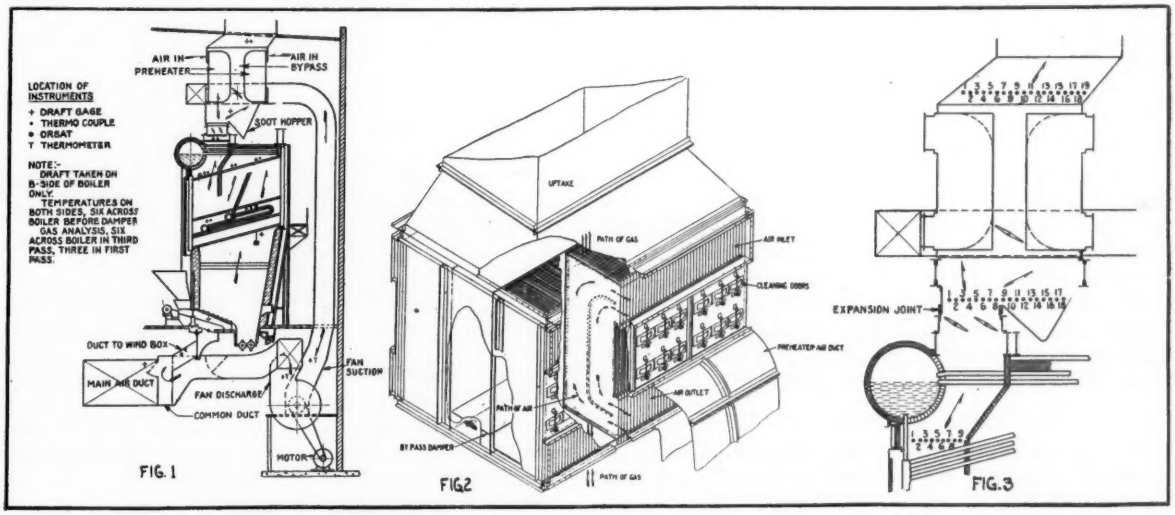


FIG. 1. GENERAL ARRANGEMENT OF BOILER AND PREHEATER SYSTEM AND LOCATION OF INSTRUMENTS. FIG. 2. DETAILS OF PREHEATER. FIG. 3. LOCATION OF CO_2 TRAVERSES AT PREHEATER

analysis of gases in the center of the first pass shows no indication of carbon monoxide. These tests were made by the Babcock & Wilcox Co. chemist and great care was made in selecting samples. The fact that no carbon monoxide was found in the first pass of course indicates that combustion is complete shortly after the gas enters the tubes.

The reduction of gas temperature through the preheater in the runs without the preheater operating, is due to air leakage and radiation, no air being drawn through the preheater by the fan during these tests.

Curves shown in Fig. 4 indicate comparative boiler performance with and without preheated air. The increase in efficiency due to preheated air as shown by the curves varies from 5½ per cent at 114 per cent rating to 7 per cent at 200 per cent rating, the general effect being a tendency to flatten out the curve in the case of the preheated air tests. This is due to the increase in heat transmission of the preheater with the increase in temperature and volume of flue gas and air through the preheater and also to the marked decrease in the per cent of combustible in the ash as indicated in the comparison of curves showing combustible in refuse.

Wind box temperature increase due to preheater varies from 120 deg. F. for 114 per cent rating to 140 deg. F. for 200 per cent rating. This higher temperature is the limit that can be reached with the preheater equipment as installed.

Exit gas temperatures are lowered in the case of tests with preheated air, the amount of reduction varying from 15 deg. F. at the lowest rating to 10 deg. F. at 200 per cent of rating. The reduction is explained by the higher mean gas temperature through the boiler resulting in more effective heat transfer. This increase in mean gas temperature is due to the decrease in gas quantity as indicated by the increased CO_2 and the higher furnace temperatures using preheated air.

Preheated Air Increases CO_2

The curves indicate an average increase of CO_2 for the results with preheated air varying from 0.7 per cent at the low rating to 1.4 per cent at 200 per cent rating. This CO_2 increase is due to better combustion conditions. The fuel bed was much more uniform and there was an almost entire absence of air holes, or disturbances of any kind to break up the uniformity of stoker operation.

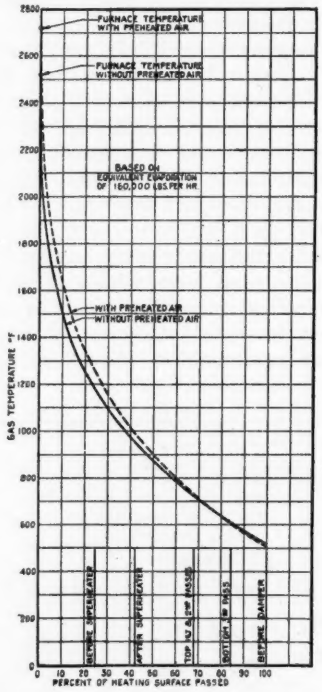


FIG. 7. COMPARATIVE TEMPERATURE DROP THROUGH BOILERS

It will be noted that the per cent of combustible in the ash remains practically constant for all ratings when preheated air is used, the decrease of per cent of combustible over that for operation without preheated air varying from 6 per cent to 12 per cent.

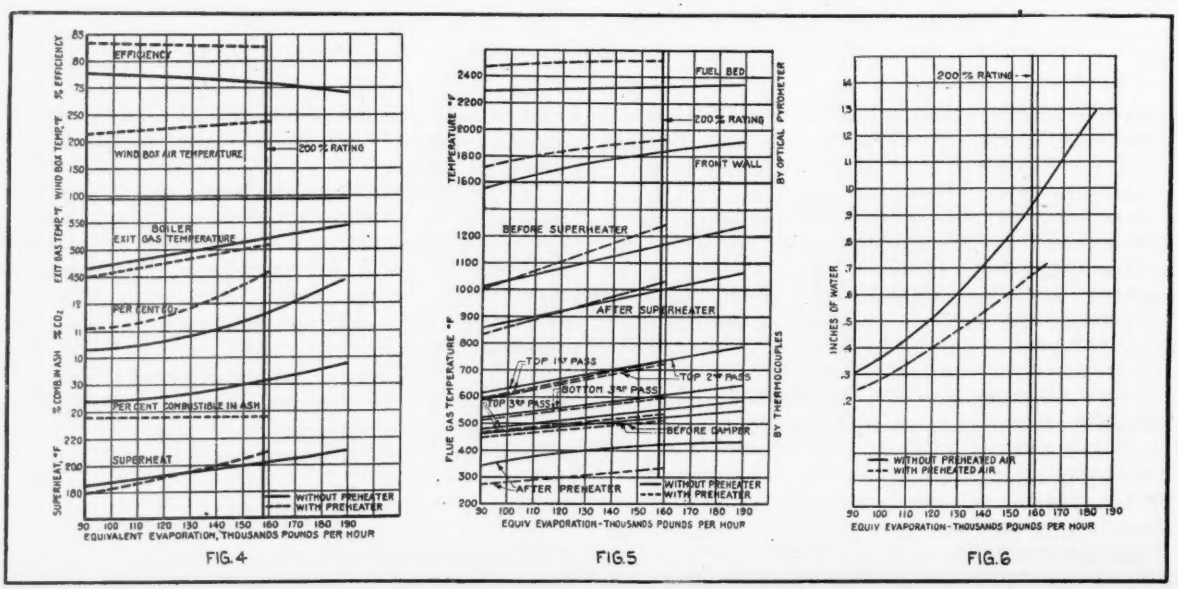


FIG. 4. COMPARATIVE PERFORMANCE OF BOILER WITH AND WITHOUT PREHEATED AIR. FIG. 5. FURNACE AND GAS TEMPERATURES WITH AND WITHOUT PREHEATED AIR. FIG. 6. BOILER DRAFT LOSSES

No material change in superheat is indicated.

The increase in fuel bed and front wall temperature for the test with preheated air over that without, shown on curves in Fig. 5, is considerably greater than the increase of the wind box air temperature on account of the reduction in excess air. The remaining curves in Fig. 5 are interesting in that they indicate the relative gas temperatures through the boiler.

Curves shown in Fig. 6 show a decrease in draft loss through the boiler for preheated air operation varying from about 20 per cent reduction for the low rating to 30 per cent for 200 per cent rating.

Figure 7 indicates the temperature drop through the boiler with and without preheated air for conditions which obtain at 200 per cent of rating. They clearly show that the mean temperature for the total travel of gases is higher with preheated air even though the final exit temperature is lower.

During the tests grinding out of the ash pit was less frequently required in the case of tests using preheated air.

Curves in Fig. 8 show special CO_2 traverses at the top of the third pass and in the uptake before and after the preheater. Such traverses were taken once for each test. The curve represents a typical example of the conditions existing. The decrease in CO_2 between traverse in the uptake before the preheater is due to air leakage at the expansion joint between the boiler damper frame and the uptake. The leakage in the preheater causes another drop in CO_2.

Figure 9 shows the approximate distribution of difference in efficiencies with and without preheated air plotted from cumulative heat balance curves with the difference in the various losses superimposed on the efficiency curves. These curves show clearly the proportion of the total efficiency gain which is due to better combustion conditions. The heat recovered from the ash pit loss is represented by the difference between curves EF and CD but the variation in radiation and unaccounted for loss brings the net gain to the efficiencies represented by curve AB.

The outstanding advantages of the flue gas air preheater as evidenced by the installation at Colfax in addition to the gain in boiler efficiencies as presented in the test results are as follows:

1. Simplicity.
2. Low cost of installation.

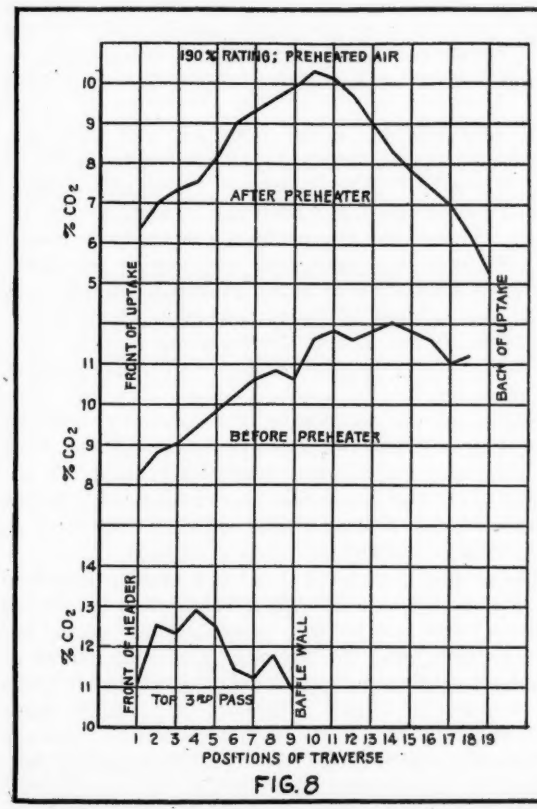

190% RATING; PREHEATED AIR
AFTER PREHEATER
BEFORE PREHEATER
FRONT OF UPTAKE
BACK OF UPTAKE
FRONT OF HEADER
BAFFLE WALL
TOP 3RD PASS
% CO2
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19
POSITIONS OF TRAVERSE

FIG. 8

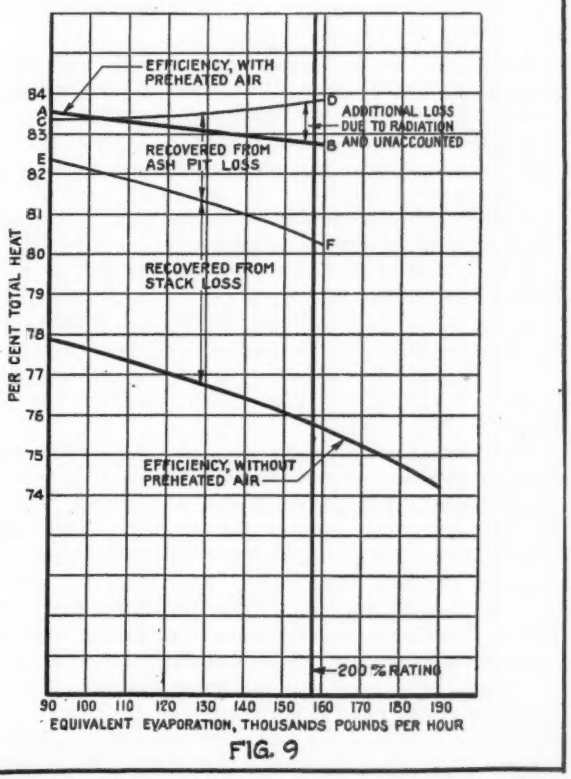

EFFICIENCY, WITH PREHEATED AIR
ADDITIONAL LOSS DUE TO RADIATION AND UNACCOUNTED
RECOVERED FROM ASH PIT LOSS
RECOVERED FROM STACK LOSS
EFFICIENCY, WITHOUT PREHEATED AIR
200% RATING
PER CENT TOTAL HEAT
90 100 110 120 130 140 150 160 170 180 190
EQUIVALENT EVAPORATION, THOUSANDS POUNDS PER HOUR

FIG. 9

FIG. 8. CO_2 TRAVERSES BEFORE AND AFTER PREHEATER FIG. 9. DISTRIBUTION OF DIFFERENCE IN EFFICIENCIES

3. Low maintenance.
4. Small operating charges.

As has been previously pointed out, the preheater equipment can in many cases at least be installed without any additional building space whatsoever.

The comparatively low cost of the equipment itself is evident from the fact that the total weight of material required to produce the same effect with water economizers is considerably greater and requires the use of an induced draft fan for the flue gas and a special pumping system for the feed water.

In the matter of maintenance, it may be argued that the preheater installation at Colfax has not been operated long enough definitely to establish reliable data. It is evident, however, that the only item questionable is the preheater element construction. There has been, up to the present time, absolutely no indication of de-

terioration in this connection and in addition the reduced amount of equipment required for the preheater installation in comparison with an economizer installation naturally reduces maintenance charges.

Operating Expense Chargeable to Preheater

The only operating cost chargeable to the preheater installation is the power required for moving the air through the preheater and duct system. In this particular installation, this amounts to about 5 kw. per ton of coal fired or the equivalent of less thon 0.4 per cent of the coal used in the boiler.

In addition to this, the more even furnace conditions resulting in a less troublesome and smoother operation, constitute an asset that is quite real even though it may be difficult to capitalize its value. In this connection the notes made by the Westinghouse stoker engineers, covering conditions observed during these tests include the following items:

1. The fuel bed was much more uniform with preheated air and no holes or disturbances of any kind appeared to break up the uniformity of operation.
2. The fuel ignited more readily.
3. The fuel bed burned much more uniformly throughout its depth.
4. There was less combustible ejected from the underfeed section to the clinker grinder pit.
5. Less carbon was contained in the refuse discharged into the ash pit.
6. The refuse in the clinker grinder pit ground out with less trouble due to the character of the clinker.
7. No clinker trouble of a serious nature was encountered at any time. The clinker itself was harder and more easily handled.

This preheater equipment was placed in operation on May 7 and was in service until June 13, when the boiler was shut down for overhauling prior to the tests. Some readjustments were made in the spacing of elements which had been forced out of line on account of uneven expansion stresses. The boiler and preheater were placed on the line again July 10 and with the exception of the period for the tests run without preheater, were in continuous operation supplying preheated air until Oct. 19, when trouble with the preheater fan necessitated taking air again from the regular forced draft fan duct. Careful inspection of the preheater revealed no indication of soot or other deposits so that it was unnecessary to make use of any soot blowing equipment, although the installation of such equipment was considered at the time the equipment was put in service. The vertical position of the elements and the weaving of the plates due to temperature and pressure variations together with the scouring action of the gases are evidently responsible for this condition.

These tests do not indicate in any sense the limit in temperature which can be successfully used for combustion air. There are no indications whatever of any undesirable effects on the stoker parts or of general combustion conditions which are not favorable to the higher air temperatures used. It is probable that still better results could be gained with somewhat higher temperature than that used.

Reduction in boiler draft loss in a measure offsets the draft loss through the preheater so that the height of stack to offset the drop in flue gas temperature is not so great as would ordinarily be expected.

Federated Engineers Urge Forest Conservation

CONSTRUCTIVE CO-OPERATION between the Federal Government and the states was urged by the Executive Board of the American Engineering Council of the Federated American Engineering Societies at its closing session in Rochester, N. Y. Respecting Federal policy, the Board adopted a resolution prepared by its Reforestration Committee, of which Charles H. MacDowell, of Chicago, is chairman, as follows:

"Pending the development of a definite co-operating forest policy on the part of the Federal Government and the several states, the Federated American Engineering Societies respectfully recommend and urge the President, the Congress, and the Federal Budget Bureau to renew yearly appropriation at the rate of $2,000,000 for the purchase of forest lands on the water sheds of navigable rivers under the Weeks Law.

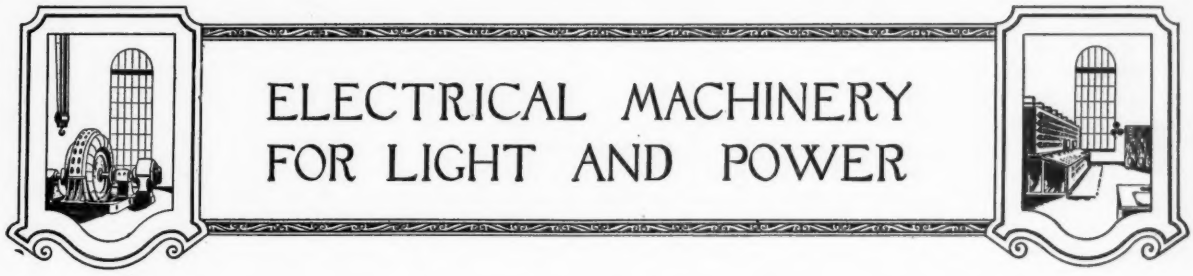
"It also feels that as fire protection is so important, present appropriations should be materially increased for co-operation with the states and individuals in fighting and preventing forest fires, and in maintaining fire patrols. The engineering profession is vitally interested in maintaining and increasing our supply of timber and forest products near heavy consuming centers, and in the maintenance of forest protection of our water sheds that future power developments may be safeguarded, river navigation insured, and valley agriculture protected."

Repeal of the constitutional provision prohibiting the cutting of timber in the forest reserves of New York was advocated in another resolution passed by the Board. The time has come, it was asserted, when a great volume of ripened timber should be cut both in the interest of conservation and of industry.

The profession of forestry is being rapidly developed and modern forest methods are well known in the United States. Trees can be cut, new plantings made, fire losses reduced, and the life and producing power of the forests continued almost indefinitely if timber tracts are intelligently treated. Further, they can be made self-supporting, and made to yield increasing revenue if rationally regarded, and this without impairing their aesthetic or recreational value.

The state, it is believed, can safely put its forest problems in the keeping of trained foresters whose reputation depends on so maintaining the forests that they will become an increasingly valuable asset, rather than a serious burden, as they are today.

PIPING SYSTEMS in power houses operating with high pressure superheated steam are usually designed with the sole object of producing an efficient system which will have the longest possible life and the least liability of a breakdown. Granting that a system constructed in the best possible manner will cost more initially than another which may operate satisfactorily, the increased investment should be looked upon as an insurance fund to guarantee continuity of operation and decreased cost of upkeep. A power plant operating under high or low pressure saturated steam does not require material of such a high safety factor as one operating with high pressure superheated steams.



Fitting the Electric Motor to the Pump

A Practical Discussion on the Application of Electric Drive to a Fundamental Operation in Industrial Plants. By R. H. Rogers*

PUMPING is a fundamental operation in all industrial plants which lends itself admirably to electric drive. The selection of pumps, motors and control for the most successful operation under all conditions and circumstances is not a simple problem and if not well studied out, the plant is penalized continuously through the following years. The fact that some of the major industries with well organized engineering departments are operating many pumps with characteristics entirely unfitted for the service would indicate that this branch of engineering needs some explaining, especially for those who have little time for an intensive study of these problems.

Perhaps the clearest way to show all the elements that affect the over-all efficiency of electric pumping units is to analyze the whole span from the electric motor to the work accomplished, not overlooking other items that make an installation successful. The items that directly affect the kilowatt-hours per million gallon feet are:

(a) Ratio of dynamic head to static head. The vertical tape measurement from level to level or the calculated head if pumping against say boiler pressure is not the measure of the pump's work. Every foot of pipe, every valve, every elbow or other fitting adds to the theoretical to make the actual or dynamic head.

For instance, considering only pipe diameter, a 5-in. three-stage centrifugal delivering 400 g.p.m. through 300 ft. of 5-in. pipe will cost $375 per year more to operate than if a 10-in. pipe were used. This differential would pay for the larger pipe in a year. Tables of heads to be added for sizes of pipes, valves, elbows, etc., at various velocities are available in pump catalogs and hand books and they deserve close consideration.

In general, the piping should have much greater area than the pump openings as high velocity is essential in the pump but detrimental in piping. Six to eight feet per second should not be exceeded and less is desirable, for, once installed, the loss goes on for years. The suction pipe especially should be deeply immersed, should be as short as is practical consistent with good installation practice and the bends, if any, should be of long radius.

An excess of dynamic head over static head means just that much more work for the motor forever after and if it is 10 per cent or 50 per cent that excess will show up at the meter.

Figure 4 shows pump installations with no suction lift as they are placed alongside the sulphite stock tank. The piping is very liberal in size so that the difference between dynamic and static head is slight.

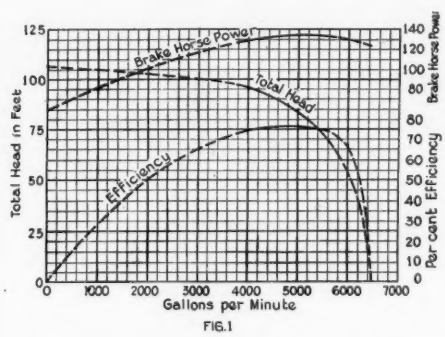


FIG. 1

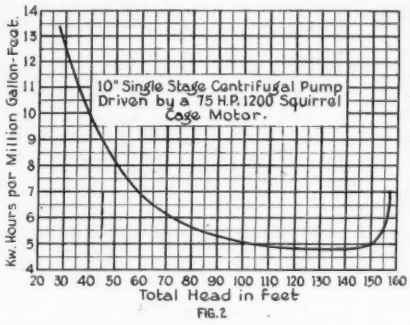


FIG. 2

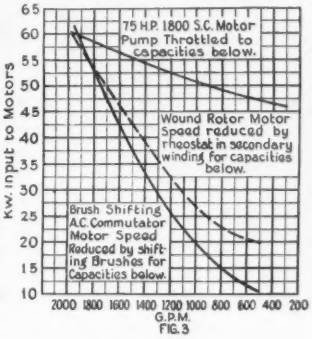


FIG. 3

FIGS. 1 TO 3. VARIOUS CURVES SHOWING PUMP AND MOTOR PERFORMANCE

Fig. 1. Typical curves for a 12-in. centrifugal pump operating at constant speed. Fig. 2. Kilowatt-hours per million gallon feet at various heads for a 10-in. centrifugal pump driven at constant speed. Fig. 3. Input to motor when reduced pump output is secured by throttling at constant speed; by reducing speed, using a wound rotor motor; by reduced speed, using a brush shifting motor.

Pump Characteristics

(b) Head, gallons per minute, efficiency, revolutions per minute and horsepower plotted together show the character of a centrifugal pump. If the speed at the point of best efficiency is at or near an induction motor

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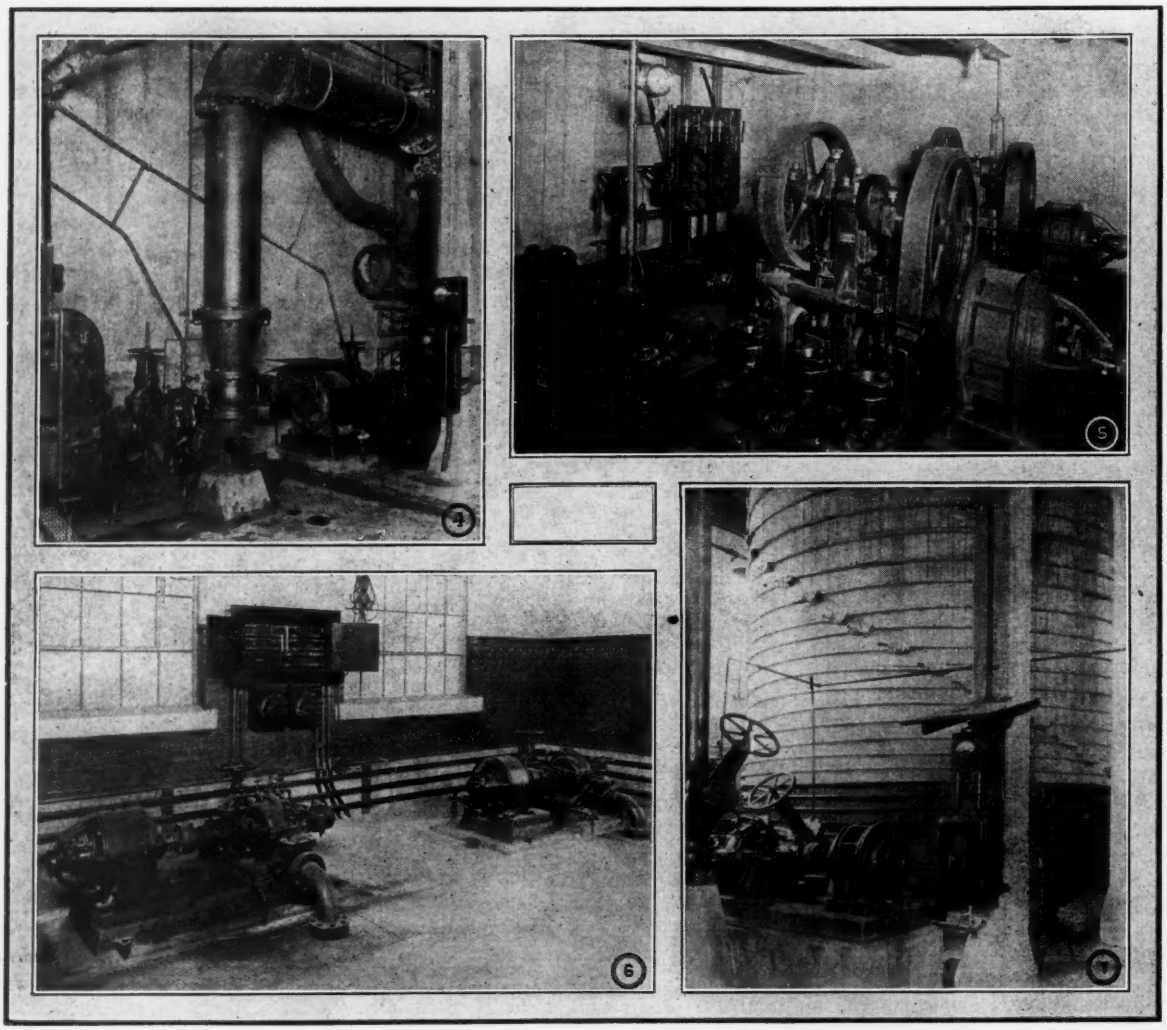
speed, viz.: 1730, 1150, 865, 690 or 575, it will insure its operation at the most efficient point if direct connected to a squirrel cage induction motor.

If the horsepower required at the most efficient point is near one of the standard squirrel cage motor ratings, viz.: 200 hp., 150 hp., 125 hp., 100 hp., 75 hp., 60 hp., 50 hp., 40 hp., 30 hp., 25 hp., etc., it will insure a high overall efficiency for the set. Fortunately induction motors have a very flat efficiency curve through the working range such as 90 per cent at full load; 90.5 per cent at 75 per cent load, and 90 per cent at 50 per cent load so that this feature has little effect. However, the power factors of such motors require that they be operated at near full load to keep the current to a minimum. Power factor affects only the leads and generating system and not the meter except as the rates may be adjusted to penalize low power factor in a plant. At loads as above, the power factor may be in the order of 89 per cent, 87 per cent and 80 per cent.

A desirable characteristic in centrifugal pumps is that of having the load fall off each way from the best operating point. This prevents an overload coming on the motor by an accident or mis-handling of valves which would subject it to maximum head (closed off) or minimum head (direct discharge).

Figure 1 shows typical centrifugal pump curves for constant speed. The efficiency is high and fairly flat from 70 ft. to 97 ft. head and from 4000 to 5600 g.p.m., hence the pump could be efficiently applied between these limits. The horsepower falls off both ways so that the motor cannot be overloaded.



FIGS. 4 TO 7. VIEWS OF TYPICAL ELECTRICALLY DRIVEN PUMP INSTALLATIONS

Fig. 4. Sulphite stock pumps in a paper mill installed to take full advantage of flexibility of electric motor drive. Fig. 5. Reciprocating pumps require the high starting torque of wound rotor motors. Float switch control at left. Fig. 6. Compound wound motors driving pumps in a nitrate plant. Fig. 7. A 40-hp. squirrel cage induction motor driving an acid pump.

Operating Off Rating

(c) Pumps are rated naturally at their best operating point and any deviation in speed, head or delivery will result in a higher cost per unit of duty.

While some future conditions may have to be given consideration, operating at lower than rated head should

be avoided especially where a constant speed motor is used. Figure 2 shows the variations in kw.-hr. per million gallon feet pumped by a 10-in. centrifugal pump, direct connected to the constant speed motor which gives the highest overall efficiency at the pump rating. The gallon-feet pumped in a year at 135 ft. head for $9100 (2c per kw.-hr.) would cost $17,000 if pumped at 45 ft. head with pumps of this class. This difference of $7900 yer year is not out of line with many pumping duty costs that may be found among the misapplied pumps in industrial plants.

Choice of Motor

(d) For constant speed service either induction motors of the squirrel cage type or synchronous motors may be used. Synchronous motors improve the power factor for the plant and this item may be enough to justify the valve manipulation incident to starting centrifugal pumps with such motors, together with the complication of bringing direct current excitation to the motor. There is little difference in the efficiencies of the motors mentioned.

For direct current service the shunt wound motor is not well adapted for centrifugal pump drive as they are too sensitive to load and voltage changes. A compound wound motor should be used with only a moderate series field say 10 per cent and such motors have been standardized for centrifugal pump service.

For adjustable speed pump drives, wound rotor motors, or the brush shifting, commutator type of motors may be used. The latter have much higher efficiencies at reduced speeds and this justifies their higher cost if operated at below normal speed frequently for long periods. Figure 3 shows the relative outlay in kw.-hr. for an adjustable speed pump using wound rotor and brush shifting motors. The third curve shows the high cost of throttling to get reduced output.

It will be noted that at 60 per cent of normal pump output, the brush shifting motor would operate 24 hr. for $10.72; the wound rotor motor for $14.88; the squirrel cage with throttled pump for $25.20. The differences become greater upon reducing the output still more.

Vertical motors may often be used to great advantage by locating the pump at whatever depth may be necessary below the motor room floor. This reduces or eliminates the suction lift with consequent benefit to the overall efficiency. Where grit is present in the water or the pump is inaccessible the entire weight of the revolving parts may be carried on a suspension thrust bearing at the top of the vertical motor. This allows of easy inspection and maintenance. Vertical motors are available in synchronous, squirrel cage, wound rotor, brush shifting and direct current construction.

Squirrel cage motors can be used on centrifugal pumps up to 500 hp. provided the inrush of starting current is not objectionable. Wound rotor motors start with much less line disturbance and are used for that reason even when intended for constant speed duty. Synchronous motors are not usually installed below 75 hp. and are of course for constant speed duty only.

Reciprocating pumps, rotary pumps and screw pumps have heavy starting torque due to high break away torque and full load pumping duty from start. For these reasons squirrel cage motors cannot be used if line disturbance is objectionable. Synchronous motors can be used but the pump must be relieved of its load in starting so that the motor can pull into step. Wound rotor motors for alternating current and compound wound motors for direct current are best for pumps of this class as they are possessed of strong starting characteristics. The brush shifting commutator type of alternating current motor is equally good where adjustable speed operation is desired.

Pump Control

Squirrel cage motors are controlled by manual or automatic compensators to which may be attached disconnecting switches and ammeters. The automatic compensators may be actuated by a push button at some convenient point, by a float switch, by a thermostat or by a diaphragm pressure switch. Figure 4 shows a manual compensator with ammeter attached, while Fig. 5 shows a motor driven reciprocating pump equipped with float switch control.

Compensators are usually shipped by their manufacturers connected to the auto transformer tap giving the lowest torque. If the motor will not start the pump with this tap connected, the next higher voltage tap should be tried, and so on, until the tap is found that provides the required torque.

To protect three-phase motors against damage due to running single phase, an overload relay should be provided. If one phase opens, sufficient additional current will be drawn through the others to operate the relay which will open the circuit to the compensator.

Low voltage protection can also be provided on starting compensators. When a condition of low or no voltage exists on the line, the low voltage protection coil is de-energized, causing the compensator handle to return to the starting position. In any installation where there is danger of a voltage shut down, undervoltage release should be specified. Undervoltage release must also be provided if the motor is to be started by a float switch or other auxiliary circuit closing divice.

The presence of explosive gases or of especially corrosive acid fumes necessitates the submergence of contact making parts in oil. Such control items are available for all classes of pump service.

Figure 7 shows an acid pump being driven by a 40-hp. squirrel cage motor, which is in no way injured by the fumes as there are no vital parts exposed. This motor is controlled by a standard compensator which has its contacts submerged in oil.

Pump Service

In conclusion it is to be noted that motor driven pumps are successfully coping with every exacting class of service in industrial plants. Liquids whether thick, light, volatile, acid or what-not are all being handled under manual or automatic control with the greatest facility. The flexibility of electric drive allows the plant engineer to draw a straight line from level to level, put the pump in the line practically without bends, with the assurance that he can bring the power to the pump without any loss in efficiency or inconvenience to the plant.

The power factor of an induction motor is about 87 per cent when operated at full load. This decreases as the load decreases, becoming of very low value at no load.

Synchronous Motors Relieve Standby Generators

RECORDS FROM PLANT SHOW BOOSTING POWER FACTOR INCREASES CAPACITY 50 PER CENT. BY H. D. FISHER

LARGE power companies are much alive to the possibilities of increasing station and transmission line capacity by power factor control and most of their power contracts for any appreciable amount of power require the consumer to proportion his apparatus so that his load will allow of maintaining 80 or 85 per cent power factor or else pay an increased rate proportional to the generating facilities he is monopolizing. Many small plant owners, however, do not appreciate the importance of such control and often increase station facilities at considerable expense when it would be possible to obtain the same result by replacing one or two induction motors with synchronous motors at comparatively little cost for the change.

Two mines in central Pennsylvania owned by the coal company with which the writer is connected, are operated by power generated in one plant which supplies the nearer mine at 2300 v. direct from the station bus and the more remote mine through a 5-mi. transmission line at 13,200-v., 3-phase, 60 cycles, with three 667-kv.a. single-phase transformers at each end. The mine supplied by the transmission line has synchronous motor driven generator sets for supplying 250-v. direct current for underground haulage and during the day when these are consuming a reasonable amount of power maintain a power factor of approximately unity at their substation; but at night, when only one of these is run and the total power consumption at that end of the line is less than 100 kw., the power factor at the generating station falls to a low figure.

Normally one General Electric turbine, rated at 2500-kw. at 80 per cent power factor, carries the full load without difficulty as it averages about 2200 to 2300 kw. with occasional short peaks at 3000 or 3200. The spare units comprise the original equipment of the station and are two 400-kw. cross-compound, high-speed Corliss engines designed for unity power factor and one 750-kw. mixed-pressure General Electric turbine designed for 80 per cent power factor.

During the summer of 1922, the large turbine lost a couple of blades and while a temporary balance was made, it was necessary to reblade part of this row and as some of the others seemed badly cut, it was decided to reblade two more rows. The material for this was received early in September and as the mines were operating at low capacity, due to labor disturbance, it was decided to undertake the work at once.

It was generally known that the engines, operating in parallel with the small turbine, would not carry more than about 300 kw.; but this had always been blamed on the condition of the engines. This did not seem reasonable, as an inspection showed them apparently in good condition and they ran easily and silently without signs of laboring. Indicator cards taken after slight adjustments to equalize cutoff showed the valve setting was satisfactory and that they were operating at about 30 per cent cutoff; nevertheless when an attempt was made to carry more than this load, the exciter voltage went to 150, the voltage regulator stopped working and alternating current voltage dropped, although all field resistance was cut out on all three machines. Switchboard instruments were checked with portable standards and found correct, so several series of readings of all the switchboard instruments were taken at short intervals, owing to the rapidly fluctuating load, to see just what the individual units were doing, and are tabulated here.

SWITCHBOARD METER READINGS AND CALCULATED POWER FACTOR

Date Test	Sept. 13,	Sept. 13,	Sept. 19, 9:09-9:39 night	Feb 13,
Time	a. m.	p. m.		a. m.
A.c. voltage	2285	2287	2335	2331
Total a.c. amperes	474	466	355	549
Total kilowatts	1412	1336	918	1993
Power factor indicator	0.745	0.735	0.653	0.910
Power factor (calculated)	0.752	0.724	0.640	0.900
Frequency indicator	60.4	60.6	61.3	61.0
Exciter voltage	130	131	137	120
No. 1 Engine-wattmeter	285	275		451
A.c. ammeter	130	123		147
Amperes-field	159	154		157
Power factor-(cal.)	0.552	0.563		0.760
Field excitation (kw)	20.7	20.2		18.8
Field excitation, per cent	7.27	7.35		4.17
No. 2 Engine-wattmeter	280	274	351	465
A.c. ammeter	131	127	140	147
Amperes-field	161	159	171	160
Power factor (cal.)	0.538	0.544	0.619	0.783
Field excitation (kw.)	20.7	20.2		18.8
Field excitation, per cent	7.48	7.62	6.68	4.13
750 kw. turbine wattmeter	847	787	567	1077
A.c. ammeter	236	215	205	272
Amperes-field	119	116	125	109
Power factor (cal.)	0.904	0.922	0.685	0.983
Field excitation (kw.)	15.5	15.2	17.15	13.1
Field excitation, per cent	1.83	1.93	3.03	1.22

It had been recognized that the field excitation on the engines was too high; but the cause of this was seen on figuring the individual power factors of the units, which showed that there was an excessive circulation of current between the generators. Various adjustments of field rheostats were tried without reducing this appreciably, but fortunately it was possible to complete the turbine repairs without the load becoming too heavy to be carried in this way.

The situation was kept in mind, however, and as mining operations increased the management was persuaded to replace a 200-kw. rotary converter at one of the substations, which was badly overloaded, with a 500-kw. synchronous motor generator set. This has proved an excellent investment as on two occasions since when it was necessary to shut down the 2500-kw. turbine for repairs, it has, by considerably over exciting the synchronous motor field, brought the power factor of the station up to 93 per cent and allowed the spare units to carry loads up to 2200-kw. average for an hour. When doing this, the engines show 550 to 600-kw. on their indicating wattmeters and are cutting off at 60 to 70 per cent stroke. The last column in the tabulation gives the average readings over a half hour which were taken after the synchronous motor was installed and shows clearly what effect the motor had in increasing the load capacity of the plant.

From this it will be seen that practically 50 per cent greater load was carried with about 7 per cent less exciter voltage, and that with the characteristics of the load the same; and whereas, in September, 1922, the voltage regulator frequently raised the exciter voltage to its limit of 155 v. and then stopped working, in February, 1923, the exciter voltage never exceeded 140 and a uniform alternating current voltage was maintained.

In this case, the increased cost of the synchronous motor generator set over a rotary converter and transformers of the same size was less than 10 per cent and, while the efficiency of the motor generator set is lower than a rotary converter, it has raised the efficiency of the distribution system as a whole and saved the cost of installing another unit in the power house.

Connecting Banks of Transformers in Multiple

By Philip G. Bernholz

ORDINARILY connecting up transformers seems to be a very simple matter to the average electrician, yet an electrical engineer with considerable experience around transformers will go slowly and make sure that the connections are correct by testing every circuit before throwing a bank of transformers on the line in multiple with another bank of transformers. Where transformers are all of the same make, they are likely to be all of the same polarity; at least, that is one of the most important tests made on transformers.

Polarity, of course, is of not much account where single transformers are connected to reduce the voltage on a line for lights or motors and where phase rotation or phase displacement are of no account; but such is not the case where several banks of large transformers turn the energy from turbines into high voltage transmission lines nor even for the rotary converter as used in so many railway stations, and direct current light and power systems supplied from alternating current transmission lines where two or three-phase alternating current is split up into six-phase to feed the rotary converters.

In this article it is my object mainly to call attention to the difficulties encountered connecting transformers with a few sketches of the most common connections and a short explanation on the reason of phase rotation and phase displacement.

Phase Rotation

Although transformers do not visibly move or rotate like a machine they are nevertheless much similar in their action. For instance, in an induction motor the stator winding is such that when connected to a three-phase circuit, there is a phase rotation that sets up a magnetic rotation which tends to pull the roter along with it in the same direction. Now, the generator that supplies the current to the motor has a similar phase rotation or, if it has not as is the case where two-phase generators drive three-phase motors or vice versa, the intervening transformers change the number of phases as well as the phase displacement; for a two-phase machine has a displacement of 90 deg., while a three-phase machine has a displacement of 120 deg. Rotary converters generally receive their energy from three-phase lines coming into the sub-station at high voltage which is stepped down and split into six-phase before entering the machine for further conversion into direct current. In the latter case, the six-phase is composed of two three-phase circuits displaced by 60 deg. from each other and here phase rotation becomes important, as both phases must rotate in the same direction as well as have the proper displacement.

Where a single induction motor is connected to a three-phase circuit, nothing can happen except running the machine in the wrong direction; but where two such machines are rigidly connected together, phase rotation in their stator windings would become important and two or more banks of transformers can be considered the same as two or more induction motors coupled rigidly to one shaft with a phase rotation equal to that of a two-pole machine or one r.p.m. per cycle.

In a single-phase transformer, the secondary current is either in step with the primary or directly opposite to it, while in a three-phase transformer the secondary may be in step or it may be displaced by 30, 60 or 90 deg., etc., according to the way the transformer is connected up. It is at first a little puzzling to see where such phase displacement comes in or why there are so many combinations and so many possibilities to make wrong connections on what seems to be such a common and apparently simple electrical apparatus.

Representation of Electromotive Forces in Transformer Windings

As the terminal voltages of a three-phase transformer are the results of two or more electromotive forces acting at different angles, the magnitude as well as the direction or angle is easily computed by drawing lines representing the direction of every electromotive force produced within the transformer as well as the magnitude of each force. Every electromotive force in a coil of wire on a certain leg of a transformer induces an electromotive force in exactly the opposite direction (180 deg.) in any coil of wire on the same leg. Therefore, in representing the two forces of primary and secondary on the same leg of any transformer, the lines must always be drawn parallel to each other, see Fig. 1 (A). With this in mind, examine Fig. 1 (B), representing star connection on one side and delta on the other side of a 1 to 1 ratio and notice that all three lines that form the delta are parallel to the three that form the star and also of the same length but the resulting voltage is not the same and there is a displacement of 30 deg. between the star and the delta voltages caused by combining the electromotive forces differently in the one case than in the other. In the same way, the displacement can be made 90 degrees as shown in Fig. 1 (C), or if each leg of the transformers has two or more coils as is generally the case, two separate deltas can be formed, one 30 deg. leading and the other 30 deg. lagging with respect to the star giving a displacement of 60 deg. between the two deltas or six-phase as used for rotary converters. See Fig. 1 (D). A more common method, however, for getting six-phase is to connect for two stars directly opposite, as in Fig. 1 (E). (F) shows one method of changing two-phase to three-phase, while (G) shows a few connections with one of the coils reversed.

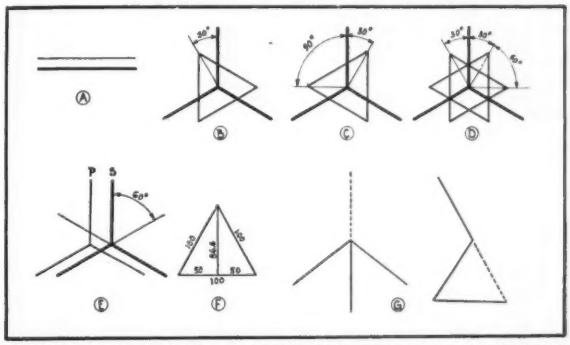


DIAGRAM ILLUSTRATING VARIOUS METHODS OF CONNECTING TRANSFORMERS

Where phase rotation is to be checked quite often a phase rotating meter is the simplest method to use. Such a meter is extremely simple to make, generally

consisting of a laminated ring of iron on which three coils of wire are wound approximately 120 deg. apart, connected in delta with three leads brought out marked A, B, C. This ring is mounted flat on a board with a compass needle pivoted in the center. When the coils are slightly excited by the machine or transformers to which they are connected, the needle will tend to follow the revolving magnetism in the ring. A small three-phase induction motor can be used the same way. Where such methods are not sufficient, fuses or banks of lights should be used to make sure of every phase, taking nothing for granted till it is proven to be positively right as such mistakes especially on large work are most disastrous.

Mechanical Atomization of Fuel Oil---III

HIGH EFFICIENCY IS MAINTAINED OVER A LARGE RANGE OF CAPACITY WITH MECHANICAL ATOMIZING APPARATUS. BY CLAUDE C. BROWN

AMONG THE indications of satisfactory operating conditions, when fuel oil is being burned, a light haze at the top of the stack is excellent. As long as this condition is maintained the indications are that satisfactory and complete combustion is taking place. In case it is impossible to see the top of the stack conveniently from the boiler-room floor, sighting across the third pass peep-holes is another good method of determining the amount of smoke in the gases. An electric light globe held at a peep-hole at one side of the boiler and observed through a corresponding peep-hole on the other side will serve to give a clear indication of the amount of smoke being produced. The light observed through the boiler in this way should be slightly dimmed, a perfectly clear gas not being desirable, as it indicates excess air. If the filaments of the electric light become simply dull glowing wires, too much smoke is being produced, and either more air or less fuel oil should be delivered until the filaments take on a fairly bright glow.

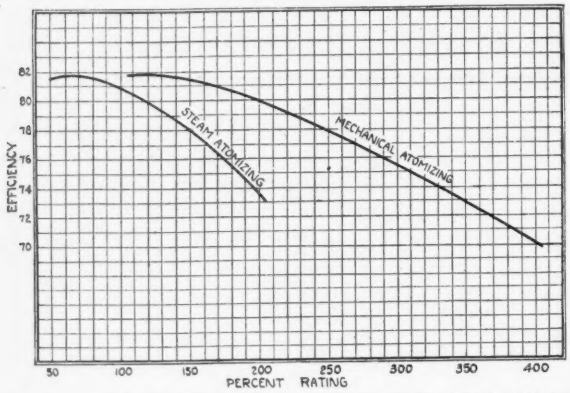


FIG. 1. COMPARATIVE TESTS OF EFFICIENCY AND CAPACITY OF STEAM AND MECHANICAL ATOMIZING OIL BURNERS

Examination of the flame in the furnace through the side peep-hole is another method of determining the quality of combustion and the relative amount of air being supplied to the furnace. If the flame is dazzling white and fairly short, with visible sparks extending beyond the flame up toward the first pass, too much air is being admitted into the furnace. If the flame is of a yellowish tinge and reaches well up into the first pass of the tube bank, practically ideal conditions of combustion have been attained. If there is a dull red tinge to the flame and wisps of black smoke are visible near the front wall, not enough air is being supplied and too much smoke is being produced. In connection with the flame examination it would be well from time to time, to examine the flame distribution at the bottom of the first pass through the peep-hole provided for that purpose. This examination will show if one burner is delivering an unusually large amount of fuel oil as compared with the others, or if one burner is smoking when the others are not. At the higher rates of driving, it may be necessary to examine this flame at the top of the first pass instead of at the bottom of it, due to the quantity of flame passing the latter point.

When all burners are working satisfactorily and a light haze or feather is present in the uptake gases the combustion in the furnace is as close to being ideal or perfect as it is probably possible to make it. Under these conditions between 13½ and 14½ per cent of carbon dioxide in the uptake gases should be readily secured and if this figure is not reached an investigation should be made to locate the cause of the trouble. It is not good practice to attempt to cut the air supply too fine or to attempt to force the CO_2 content of the gases up too high for the reason that in so doing invariably CO is formed in the furnace, with its consequent losses. In sampling the uptake gases for making determination of the percentage of CO_2, great care should be taken to obtain an average sample of the gas that will be representative of the entire gas stream and when examination is made of the sample for CO_2 additional determination should be made for carbon monoxide and hydrogen.

CARE AND INSPECTION OF BURNERS

One of the most important points in connection with the satisfactory operation of the burners of a mechanical atomizing system is the inspection and care of the atomizer. It is essential when burning fuel oil by this means that the atomizer be kept thoroughly cleaned at all times, as a slight plugging in any of the oil passages causes poor atomization at the tip; a spray streaked with black streams of unatomized oil which drips into the furnace and poor and incomplete combustion results. When an atomizer in operation does not give a perfectly uniform cone of fuel oil spray, it should be taken out and replaced by a spare atomizer and then thoroughly cleaned and placed on the spare atomizer rack. The operator should be careful to observe the tips of the atomizers (through the peep-holes provided in the top air door of the register) at frequent intervals to observe the appearance of the oil spray coming from them. Even a complete system of strainers does not entirely prevent the plugging of the atomizers. As a general thing, the present day average grade of fuel oil contains varying quantities of lint-like material that will pass any strainer and which is bound to lodge somewhere in the oil passages. For this reason, it is well to change the atom-

izers at periodic intervals even though they seem to be giving satisfactory service. This lint builds up in the passages and its presence is not noticed when driving at ordinary loads; but when the full capacity of the apparatus is needed for the higher ratings, any such plugging or congestion of the oil passages cuts down the capacity of the system.

If the cause of the plugging is discovered in the sprayer plate, it will be unnecessary to remove the nozzle; but if nothing is found there the nozzle should be thoroughly cleaned and examined. If the atomizer has been in operation for any length of time, say 12 or 15 hr., the strainer in the body of the burner will undoubtedly need cleaning. It is good practice thoroughly to blow out with steam all apparatus that has been cleaned to make sure that all of the passages are clear.

Should any carbon build up on the cast-iron impeller, on the brickwork around the burner opening, or on the furnace side walls or floor, it may be cleaned away in any one of several ways: First, by shutting off the burner which is building up the carbon and leaving it shut off until the radiant heat from the other burners and the hot brickwork burn it off. In doing this the air doors in the register of the burner that is shut off should be left open a small amount to supply the air necessary for the burning up of this carbon. Care should be taken that only sufficient air is admitted to burn off the carbon completely. Second, by cutting the carbon away with a rod or bar of iron (flattened to a chisel edge at one end) inserted through the wide open air-doors of the register.

Due to the wear and tear on the brickwork, castings, etc., by the second method, it is decidedly preferable to use the first method if the power requirements will allow a burner to be shut off long enough to burn up the carbon. Before the burner is again lighted up, a careful examination should be made to determine the cause of the formation of the carbon and this cause eliminated. The formation of the carbon may be due to one or all of a number of causes, as for example the fuel oil may be too cold for proper atomization, or one of the air doors of the register may have become stuck in a closed position, or all the air doors in the register may be too far closed so that the opening is too restricted to allow sufficient air to enter the furnace for the proper combustion of the fuel oil. The atomizer may be dirty or clogged so that the flame is not central, but is hitting the brickwork. Too much excess air may be entering the furnace chilling the furnace brickwork below the ignition point of the oil. There are a large number of conditions under which the apparatus will not operate properly and as a general thing the causes of these conditions may be traced directly to carelessness. Be careful, use good judgment, keep the apparatus clean, and the mechanical atomizer will invariably give good results.

Relation of Efficiency to Capacity

One of the principal reasons for the development of the mechanical atomizing oil burning apparatus is the fact that the mechanical systems have a greater capacity than the steam or air atomizing type and the higher boiler efficiencies which are attained only at or near the rated capacity of the apparatus in the case of the steam atomizing type are attainable at high overloads when the mechanical systems are used.

There are a number of advantages in the use of fuel oil over the use of coal among which might be mentioned: absence of ash and clinker, ease of transportation, storage, and handling, and economy of both space and personnel. If, however, fuel oil is to take the place of coal, in the furnaces and boiler plants of our industries, the oil burning equipment used must be able to carry large sustained overloads which are now carried by the coal burning plants and it must also be able to carry these overloads at high efficiencies. At and around 100 per cent rating, the steam atomizing type of burner can be made to give excellent results, but the efficiency falls off rapidly at the higher overloads, and the extreme overloads demanded in many of the large

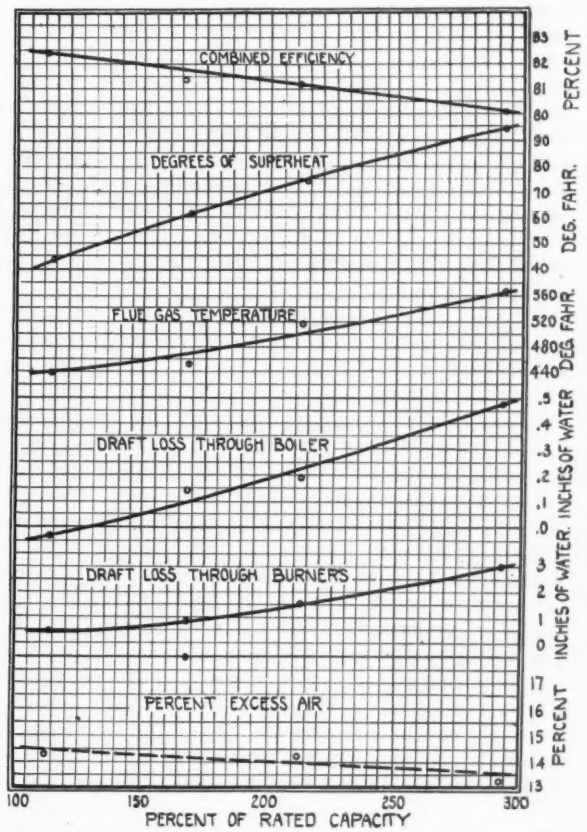


FIG. 2. PERFORMANCE CURVES OF MECHANICAL ATOMIZING BURNER

plants of today have been found impossible of attainment in the usual form of furnace with this type of apparatus.

Present day practice of operating boilers in large stations, up to 300 per cent of rating during the times of peak load has made it necessary to consider some other means of burning the fuel oil than with steam atomizing burners, as a unit that would normally be operated over peak load periods at 300 per cent of rating or thereabouts with coal, could not be operated at over 200 per cent with the steam atomizing apparatus.

Among the investigators who have made a study of this situation one developed a system of mechanical atomization and made a series of tests to determine the possibilities of the utilization of this apparatus under

stationary boilers. The tests indicated that the economy obtainable with steam atomizing and with mechanical atomizing oil burners at the lower boiler capacities was about the same. At the higher capacities and overloads, however, the efficiency obtained with the mechanical atomizing burners was higher than that attained with the steam atomizing type. Considerably higher capacities could be obtained with the mechanical atomizing burners than with the burners which used steam as an atomizing medium. The results of these comparative tests are shown in Fig. 1, where the lower curve represents the net efficiencies obtainable at the different ratings with the steam atomizing burners and the upper curve the corresponding efficiencies with the mechanical atomizing type of apparatus. By net efficiency of the steam atomizing burners is meant the efficiency based upon the total steam evaporated less the steam blown to waste through the burners in the atomization of the fuel oil. In the case of the mechanical atomizing burners, no deduction is made for the relatively small amount of steam required for the mechanical atomization of the fuel oil, as the heat in this steam is usually returned to the system by employing it for heating the fuel oil or by passing the exhaust steam from the oil pump to the feed water heater. At any rate, the heating and the pumping of the fuel oil is necessary in either case, and the results are comparable.

It will be noted from the curves in Fig. 1, that with the boilers operated at rating, the efficiency in each case is in the neighborhood of 82 per cent. At 200 per cent of rating, which for the usual type of boiler employed in large boiler plant practice, represents the approximate limit of capacity for steam atomizing burners, the difference in efficiency in favor of the mechanical atomizing burner is in the neighborhood of 6 per cent. It will be further noted that the rating-efficiency curve of the mechanical atomizing burner is flatter than that of the steam atomizer and the higher efficiency is sustained for a longer period. At an efficiency of 73.4 per cent the steam atomizing burner has a capacity of 200 per cent of rating, while at the same efficiency the mechanical atomizing burner has a capacity of 340 per cent of rating. As stated by D. S. Jacobus and N. E. Lewis, of the B. & W. Co., the curve in Fig. 1 for the efficiency secured at different ratings with steam atomizing burners is based on tests that were made on a 14 high B. & W. boiler at the Redondo plant of the Southern California Edison Co., using California fuel oil of about 14 deg. Beaumé with a heat value of about 18,000 B.t.u. per lb. The curve for the efficiencies secured at different ratings with the mechanical atomizing burner is based on tests made on a 14 high B. & W. boiler at the Bayonne works of the Babcock & Wilcox Co., with Mexican fuel oil at 14 to 16 deg. Beaumé having a heating value of approximately 18,300 B.t.u. per lb.

In the tests of the mechanical atomizing burners the conditions existing were as follows: The pressure of the fuel oil at the burner varied from 100 to 200 lb. per sq. in., depending upon the capacity at which the boiler was operated and upon the size of the sprayer or orifice plates which were used inside the tips of the mechanical atomizers. The best atomization was obtained when the fuel oil was heated to give a viscosity of from 3 to 5 deg. Engler, which required a temperature of from 200 to 250 deg. F. Live steam was used for heating the fuel oil from about 100 deg. F. to the temperature specified, the amount of steam utilized for this purpose being approximately 0.5 per cent of the total steam generated.

In the tests the amount of steam required for driving a rotary pump which was used for pumping the fuel oil amounted to from 1 to 1½ per cent of the total steam generated. The pump was considerably larger than was required and was therefore wasteful. In a large plant the amount of steam required for pumping the fuel oil would be in the neighborhood of ½ to 1 per cent of the total steam generated, and the exhaust steam from the pump could be used for the preliminary heating of the fuel oil.

Figure 2 shows a series of curves obtained from tests made on the boilers of the Riverside plant of the Savannah Electric Co. indicating remarkable results obtained with mechanical atomizing equipment. The boilers used were B. & W. 665 hp. units, 14 tubes high, 21 wide, equipped with Foster superheaters. All boilers were originally equipped with stokers for bituminous coal. Four mechanical atomizing fuel oil burners were installed under each boiler, to operate under forced draft. Mexican oil was used, heated to a temperature of about 230 deg. F. and at a pressure of approximately 210 lb. per sq. in. or lower depending upon the capacity desired.

Where a change is made from coal to fuel oil firing, the mechanical atomizing oil burners are particularly adapted, as in many cases the stokers can be protected with brickwork and the oil burned directly above the stokers without removing them. By keeping the stokers in place, the boilers can be readily converted back to coal firing if desired. The same blast that is used for forced draft stokers may be employed for the mechanical atomizing burners thereby saving the expense of a portion of the oil burning equipment.

Mechanical Burners Have High Capacity

Higher capacities are available with mechanical atomizing burners than with steam atomizing burners on account of the quicker combustion secured with the former and because the steam atomizing burners do not utilize the full volume of the furnace. With mechanical atomizing burners, the fuel oil is expelled under a heavy pressure from the burner tips in a fine mist and intimately mingled with the air for combustion directly at the burners, which leads to the greater part of the fuel oil being consumed in short cone-shaped flames. With steam atomizing burners, the fuel oil is not divided into nearly as fine a spray, and there is no quick and thorough mingling of the oil and the air, which leads to the long flames characteristic of the type. To secure good combustion with steam atomizing burners, the flames should have a natural path of travel, the air for combustion being admitted beneath the flame. It can be appreciated that only, say one-half of the furnace volume may be occupied by the flames of a steam atomizing system and that much of the furnace volume may be ineffective. With the mechanical atomizing burners there are two effects which allow a greater capacity to be developed with a given furnace volume, the first being the rapidity of combustion due to the better atomization and the more thorough mingling of the air and the atomized fuel oil at the burners, and the second, the form of the flames which makes available a greater proportion of the furnace volume. With steam atomizing burners the amount of fuel oil that can be properly burned in a furnace is about 2½ lb. per cu. ft. of total furnace volume

per hour, whereas with mechanical atomizing burners this figure may be as high as 10 lb. or more.

Draft Requirements

With a natural draft of 1 in. of water available, the capacity with mechanical atomizing burners with the number of burners that can be installed in the ordinary setting for coal firing, would be limited to approximately 200 per cent of rating. At higher capacities than 200 per cent, forced blast at the burners should be used. The number of burners for use in a given boiler may be determined on the basis of one burner for every 120 or 130 rated boiler horsepower. This capacity may be exceeded in certain cases. In operation the variation in load is taken care of by adjusting the fuel oil pressure at the pumps between the limits of 100 and 200 lb. per sq. in., and by cutting in and out burners. Throttling the fuel oil at any individual burner should not be done and the valve to a given burner should remain always wide open or tight shut. It is not advisable to attempt to regulate the air to any great extent with the air doors of the individual burners. The best and most satisfactory method is through the use of the boiler dampers for natural draft installations. Where forced draft is employed, the air is regulated through the use of the boiler dampers and the adjustment of the air blast.

Boilers fitted with mechanical atomizing burners require more draft than those fitted with steam atomizing burners, at any given rating, as the air for combustion must be drawn through the burner registers at a velocity that will cause it to mingle intimately with the atomized oil. With no forced draft, the amount of draft suction required at the damper of a 14 high B. & W. boiler for drawing the air through the burner registers and for drawing the gases through the boiler when operating at rating, 150, 200 and 250 per cent of rating, is, 0.25, 0.6, 1.0 and 1.65 in. of water respectively. Where forced draft is used at the burners, the draft suction required at the boiler damper to overcome the resistance of the gases flowing through the boiler is, 0.15, 0.20, 0.35 and 0.50 in. of water respectively and when operated at 250 per cent of rating, the forced draft required at the burners is 1.15 in. of water.

While the application of mechanical atomizing fuel oil burning equipment to the stationary boilers of central stations and isolated power plants is as yet a comparatively new development, tests that have been made that, together with results obtained at a number of installations that have been in operation for some time and where the mechanical atomizing apparatus has been commercially installed, indicate that this type of equipment meets all the conditions that have been met thus far by the coal burning equipment and does so in a much more flexible and adaptable manner than does the latter. The mechanical atomizing systems fill the gap between the coal burning system and the steam atomizing fuel oil burning systems, having all of the advantages, flexibility, economy and ease of control of the fuel oil burning system and at the same time embodying the high efficiency and high capacity for long sustained overloads that is possible with coal.

Operation of the mechanical atomizing system has been found to be extremely simple. Only two conditions require adjustment at the furnace, oil pressure and draft, and in many plants it has been found possible to schedule these conditions and their adjustments so accurately that the everyday operating performance closely approximates the best test results.

In one plant, a single size of burner tip has been selected which fulfills all requirements of normal operation. There is no adjustment of air registers for burners in operation, as they are set wide open. There are, consequently, constant orifices for the admission of both air and fuel oil. The proper relation between the air and oil pressures throughout the capacity range has been determined by analysis of the gases of combustion, and an additional scale etched upon the dial of the air pressure gage gives the corresponding oil pressure required. In operation, the oil valves are adjusted to give the oil pressure indicated on the oil scale of the air gage for whatever air pressure is being carried at the moment, and the resulting conditions are right for proper combustion. The simplicity of the fireman's work is evident. Little difficulty is experienced at this plant in maintaining smokeless conditions with varying load at all ratings.

At this same plant, an interesting outcome of the increased furnace efficiency has been the effect upon superheater performance. When burning coal, the steam was superheated on the average about 140 deg. F., and it was necessary to keep constant watch of the condition of the boiler heating surfaces to avoid exceeding this figure. With combustion entirely completed in the furnace, as occurs in the case of the mechanical atomizing oil burning apparatus, and the resulting increased heat absorption by the first pass of the boiler proper, the superheat imparted by the same area of superheater surface is reduced by approximately one-half.

Explosion Hazards of Pulverized Coal

L. D. Tracy, superintendent of the Central District Experiment Station of the Department of the Interior, Bureau of Mines, Urbana, Ill., has completed a study of explosion hazards in industrial plants attending the use of pulverized coal. This investigation was begun in 1919 because a number of explosions with consequent loss of life in industrial plants occurred soon after the introduction of pulverized fuel, and because of the bureau's previous experiments regarding the explosive qualities of coal dust. The purpose is to give the users of pulverized coal a knowledge of the hazards involved, so that by proper safeguards and careful operation, further loss of life will be avoided. The scenes of actual explosions have been visited, the different kinds of apparatus studied, the knowledge of the explosive characteristics of coal dust applied, methods of prevention of explosions considered, and rules and regulations for safe operation have been prescribed, all of which will be incorporated in a report being prepared. This information should be of direct application by the large manufacturing interests in and around Chicago and other parts of Illinois, as well as to the manufacturers in other states.

Correction Note

In the description of a new development in soot blowers which appeared on page 1212 of the Dec. 1 issue, the name of the manufacturer was inadvertently omitted. This soot blower is a development of the Bayer Co., of St. Louis, Mo.

THE INQUIRING ENGINEER

The Inquiry

Will the substitution of the internal combustion engine for the usual non-condensing or bleeder type turbine for house generator service in central stations be practical and economical? What desired conditions in the heat balance would warrant its use? What factors should be considered in an analysis of the problem? What, if any, are the advantages to be gained?

Replies

ROBERT E. DILLON, Asst. Supt. Generating Dept.,
The Edison Electric Illuminating Company of Boston.

Aside from the important consideration of saving the natural resources of the country, the prime object of the central station is to give reliable service to the customer at the least possible cost, after paying a fair return on money invested.

Attainment of this object is made possible only by reducing to a minimum the sum of the carrying and the operating charges, without impairing the reliability of service.

No method common to all proposed stations can be laid out for figuring these charges. Each installation is governed in this respect by a number of individual considerations such as size and location. The combination of the carrying charge, operating costs other than fuel, cost of maintenance and the reliability factor is, in most cases, an obstacle to the use of the internal combustion engine.

From the fuel standpoint, all combined auxiliary power and heating systems are nearly one hundred per cent efficient thermally; also, the fuel supplied for feed water heating before performing this function usually gives up more than sufficient power for auxiliary uses. Consequently, the question of fuel consumption resolves itself into one where the arrangement gives the greatest percentage of power per unit of fuel used for feed water heating. If the percentage of power so used is larger for the internal combustion engine than obtained with other arrangements, the saving effected would have to more than offset the balance which has been indicated as adverse to its use.

JOHN M. DRABELLE, Mechanical and Electrical Engineer,
Iowa Railway and Light Co., Cedar Rapids, Ia.

I do not believe that the use of internal combustion engines for house generators is practical, particularly in the large or so-called super-power plants, for the reason that the total horsepower of station auxiliaries would result in some extremely large units and it is my personal opinion that the oil engine has as yet not been developed to a point of sufficient reliability to trust the auxiliaries of a large station to it as the source of power.

Certain developments are being made, such as, for instance, the auxiliary generator direct connected to the main generating unit, that is being installed at the Waymouth station at Boston; stage heating for feed water, which surpass the gas engine idea. I cannot see any good use for an internal combustion engine in this capacity.

VERN E. ALDEN, Asst. to General Superintendent,
Consolidated Gas, Electric Light and Power Company of Baltimore.

There is unquestionably a definite need in connection with all of our large power stations that the power supply for certain of the auxiliaries shall be made as reliable as is physically possible. This need has led to the use of the house generator, driven by an independent steam turbine.

There are at least three definite trends in power station design at the present time which will, I believe, result in the elimination of the independent turbine driving the house generator. The use of higher steam pressures and temperatures makes it imperative that the amount of steam piping and the number of valves and fittings be cut down to a minimum and that just as few pieces of equipment as possible shall come directly in contact with steam at the higher pressure and temperature. The ideal which we are striving for is one boiler and one turbine connected together by means of a short length of steam piping with not more than two valves in it.

In striving after higher station economy, we are anxious to get the maximum amount of energy out of each pound of steam. It is doubtful if the house turbines used in connection with any of the large power stations operate at a Rankine cycle efficiency ratio of higher than 60 per cent. The main turbines in our large power station are operating with a Rankine cycle efficiency ratio of 75 per cent, over all, including generator losses. It is readily apparent, therefore, that we will get more energy out of a pound of steam by letting it pass through the main turbine than if we expand it through the blades of an independent house turbine. This same desire for higher over-all economy in our newer power stations has brought us to the use of the regenerative cycle, steam being bled from the main turbine at three or four different points for the purpose of heating the feed water. The over-all efficiency of the regenerative cycle with stage bleeding in connection with the main turbine is appreciably higher

than any combination of a main turbine with an independent hous e turbine of either the condensing, the noncondensing, or the bleeder type. There is now as always the pressure on the designing engineer to keep the first cost of the power station to a minimum. The elimination of the independent turbine driving the house generator and the elimination of the additional amount of piping required in connection with this separate turbine, work in this direction.

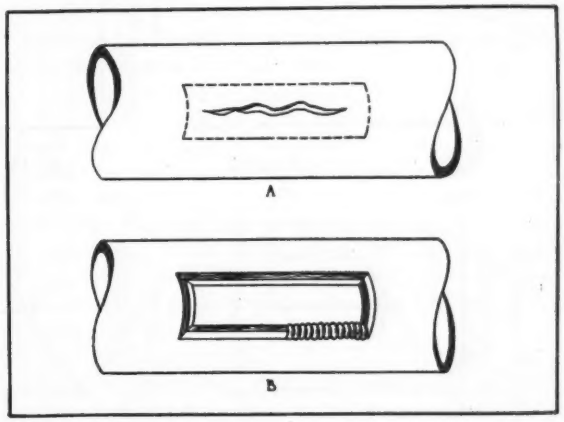
It is perfectly evident, however, that if we eliminate the independent turbine driving the house generator we must substitute something in its place that will still permit of an adequate amount of insurance in connection with the auxiliary power supply. There are two main alternatives; one is to drive the house generator by an internal combustion engine, and the second is to couple the house generator on to the end of the shaft of the main generator and drive it by means of the main turbine. The latter alternative, to me, appears to offer the greater advantages. The major consideration in connection with the selection of a prime mover for the house generator should be reliability. As central station men, we have never conceded that an isolated plant with an internal combustion engine used as a prime mover could be anything like as reliable as one of our large power stations. The power station is no more reliable than its source of auxiliary power supply, and it would in my opinion be a grave error both from an engineering and a commercial standpoint to install an internal combustion engine for furnishing power to the station auxiliaries.

With the use of the regenerative cycle by means of stage bleeding in connection with our main turbine units, it is possible to increase the output of a standard turbine between 10 and 20 per cent. This increased output is obtained by passing a large amount of steam through the first stages of the turbine. The amount of steam passing through the low pressure stages where the real limitation occurs, is not increased. We are up against the problem of finding a way to avail ourselves of this additional capacity in the steam end of the unit without entirely re-designing the main generators. It is apparent, therefore, that if we couple the house generator direct to the main generator we will be helping to solve this problem and it will not be necessary to add to the cost of the steam end of the unit in order to obtain a total output which will be the sum of the nameplate ratings of the main generator and the house generator. It is to be noted that with this layout the first cost will be comparatively small as compared to the use of an internal combustion engine and it is certain that the maintenance charges will be much lower.

One further point is to be noted. A high degree of reliability in connection with the power supply is required only in connection with certain of the power station auxiliaries as, for instance, the circulating pump, the boiler feed pump, the condensate pump and the air removal pump. In the event of trouble on the lines feeding out from the station or in the event of trouble on the station bus bars, there is no objection and possibly there is some slight advantage in a momentary shut down of the auxiliaries in the boiler room such as the forced and induced draft fans and the stoker motors. Bearing in mind that the auxiliaries may be divided into two classes and that only one of these classes needs a thoroughly reliable power supply, it becomes possible in some cases at least, to derive this reliable power supply from an extra large direct connected exciter mounted on the end of the main generator shaft. In connection with two new turbines which we are installing in our Westport Station in Baltimore, we are using a 500-kw. direct connected exciter to supply the power for the excitation of the main generator field and for driving the variable speed motors used in connection with the circulating pump, the condensate pump and the boiler feed pump. In this case, a steam jet air pump is used for removing the air from the main condenser.

Ruptured Pipe Repaired by Welding

WHEN a pipe breaks, it is usually feasible to assume that it gave way in the weakest point, and in making the repair by oxy-acetylene welding it is well to keep in mind that the metal immediately in the vicinity



PIPE REPAIRED BY REMOVING DEFECTIVE SECTION AND BUTT WELDING PATCH IN PLACE

of the break is probably weak also and for this reason it is best to remove it for a short distance back from the break.

To repair a longitudinal rupture, which is the usual class of break, the following method is recommended: With an oxy-acetylene cutting blowpipe cut away the ruptured and weakened metal leaving a rectangular hole in the pipe as indicated by dotted line in sketch A. Then a rectangular patch of the same thickness and form may be obtained by cutting it from a stock or scrap length of the same size and class of pipe or it may be readily formed from a piece of sheet metal of the same thickness as the pipe wall. If this thickness is $\frac{1}{8}$ in. or more, the edges of both the hole and the patch should be beveled to a 45-deg. angle with the cutting blowpipe so that when the piece is inserted a 90-deg. V will be formed, thus permitting the welder to obtain full penetration of the joint when he butt welds the patch in place. (A piece of rod welded to the patch will serve as a handle by which it can be held in place during welding and is readily cut off with the blowpipe after the welding is completed.) This type of repair can be completed in an hour or two, can be made at small expense, and is a permanent job.

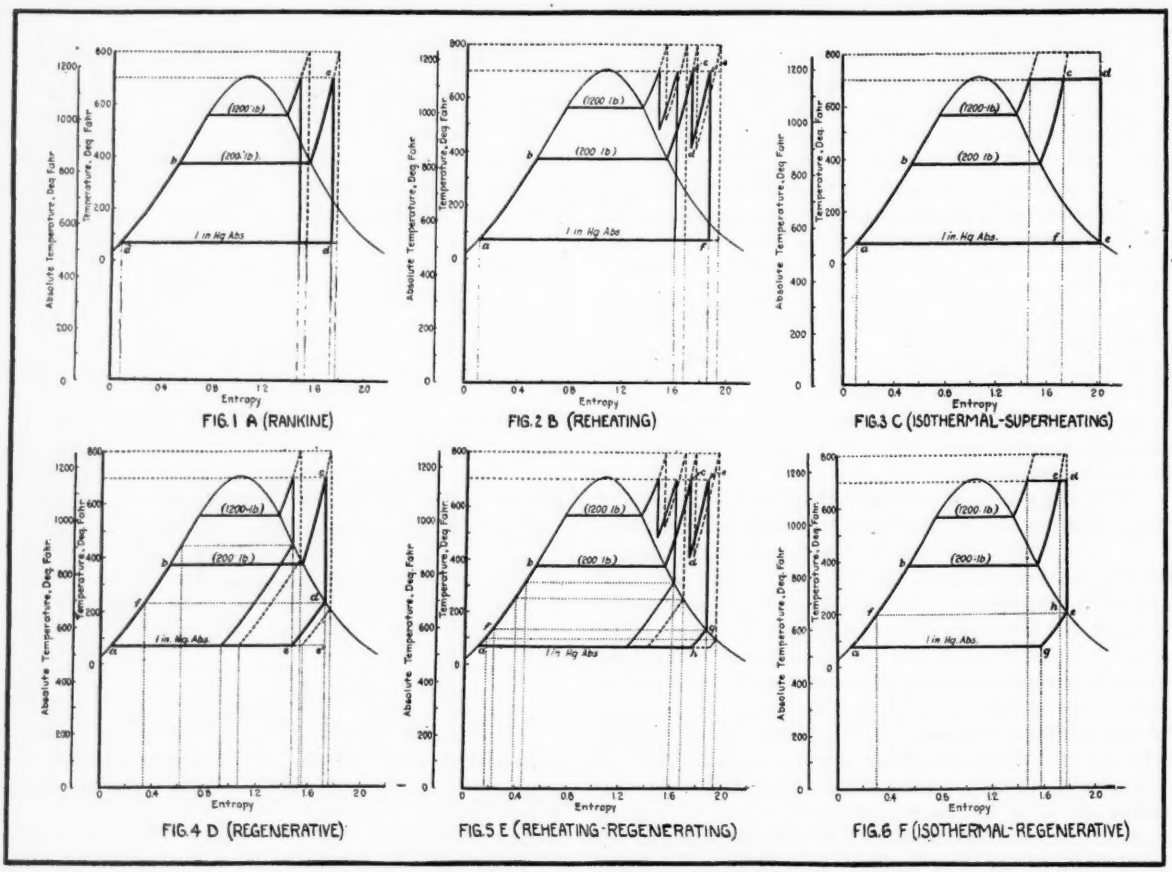
A Study of Steam Power Plant Cycles*

RELATIVE THERMAL AND INVESTMENT COSTS FOR TURBO-GENERATOR STATIONS EMPLOYING STEAM PRESSURES FROM 200 TO 1200 LB.; SIX DIFFERENT CYCLES OF OPERATION INVOLVING REHEATING AND REGENERATING. BY C. F. HIRSHFELD AND F. O. ELLENWOOD

AN IDEAL steam-power-plant cycle is one in which water and steam are assumed to pass through a series of physical processes in a perfect fashion so that the losses due to throttling, turbulence, friction, radiation, conduction, and other phenomena unavoidably present in real plants, are excluded. The only loss that remains in such a case is the thermodynamically unavoidable one due to the heat rejected in the exhaust.

Unfortunately, the actual steam plant cannot be operated under such ideal conditions. The exhaust loss in a real plant is larger than all the other losses combined, so that the best way of utilizing some of this exhaust heat, other than by industrial heating—which is sometimes feasible—becomes a problem of increasing importance as higher fuel prices are encountered.

In considering the ideal cycles for any type of heat engines, that of Carnot is always entitled to first place. This cycle is too well known and too difficult to approach in a real steam plant, to require at this time any discussion or diagram. Attention will simply be called to the fact that with the upper steam temperatures of 700 and 800 deg. F. and an exhaust temperature of 79 deg., the corresponding Carnot efficiencies are respectively 0.535 and 0.572, which are the maximum attainable efficiencies with these temperature limits.



FIGS. 1 TO 6. IDEAL STEAM-POWER-PLANT CYCLES

For throttle pressures of 1200 and 200 lb. per sq. in. abs., temperatures 700 and 800 deg. F. and exhaust pressure of 1 in. mercury abs. Fig. 1. Rankine, Cycle A. Fig. 2. Reheating, Cycle B. Fig. 3. Isothermal-superheating, Cycle C. Fig. 4. Regenerative, Cycle D. Fig. 5. Reheating-regenerating, Cycle E. Fig. 6. Isothermal-regenerative, Cycle F.

Of the six other cycles considered, the Rankine is the only one that is almost universally called by the same name throughout the engineering world. The different cycles have been designated respectively by the first six letters of the alphabet. The names chosen are intended to be characteristic of the most unusual parts of the cycle, or in other words to identify the cycle by a name which indicates the unusual processes involved. All of the cycles herein considered have certain common characteristics, namely, the constant-pressure processes represented by the lines *abc* in each figure.

The Rankine Cycle (A) is made up of the processes indicated in Fig. 1, in which the cycle is drawn for

*Abstract of paper presented at the Annual Meeting of the A. S. M. E.

initial pressures of 200 and 1200 lb. and maximum temperatures of 700 and 800 deg. F. In this cycle the steam is formed at constant pressure and is then considered to pass without any loss by throttling or radiation to the turbine, in which the adiabatic expansion, *cd*, takes place until exhaust pressure is reached as indicated by the state *d*. Condensation of the exhaust steam then takes place at constant pressure as shown by the line *da*. This cycle has much to recommend it from the standpoint of simplicity and is the one in most common use today. It does not, however, have as great possibilities regarding thermal economy as some of the others.

Reheating Cycle (B) is composed of the processes shown in Fig. 2. The steam expands adiabatically from *c* to *d* and then is reheated at constant pressure from *d* to *e*, after which a second adiabatic expansion takes place from *e* to *f*. The rest of the cycle is the same as cycle A. This reheating implies that the steam is carried back to the boiler plant to pass through a suitable reheater and is then returned to the turbine in which the expansion is completed, or else that there has been provided near the turbine a suitable separately fired reheating apparatus so that the steam does not have to be carried back to the boiler room. Both of these schemes involve certain complications and possibly difficulties in a real plant.

With Isothermal Superheating the Cycle (C) consists of the processes shown in Fig. 3. This name is used because the cycle involves an isothermal expansion of superheated steam as indicated by the line *cd*. It is not claimed that this process is likely to be approached even approximately in any real turbine, as it now seems that the expense and complication resulting from the attempt to supply heat to the steam while it is expanding in the turbine would probably more than offset the advantages to be gained thereby. The study of this cycle, however, does offer one very great advantage in that it shows a maximum which might be approached by using an infinitely large number of constant-pressure reheating devices as explained for cycle B.

The Regenerative Cycle (D) is made up of processes which may be indicated as in Fig. 4, although some of these processes are complex. This important cycle is called "regenerative" because steam is bled from the turbine at a number of stages to feed water heaters, and the heat which is thus recovered is returned to the boiler.

In the ideal cycle, an infinitely large number of bleeder heaters are assumed to be used, so that if superheated steam is not bled, the line *de* is drawn parallel to the liquid line *fa*. If there had been no bleeding, the adiabatic expansion would have continued to the state e′. On the other hand, with bleeding, successive portions of the steam are withdrawn from the turbine and condensed, heating the feed water, while the remainder proceeds through the turbine, continuing the isoentropic expansion.

If at each temperature the portion in the turbine and the portion which has been bled and condensed be considered to be mixed together, the resultant state is that shown on a curve drawn parallel to the liquid line. In other words, the curve *de* is drawn for the entire weight of steam considered in the cycle and does not represent the state of only that portion of the steam continuing through the turbine. By drawing the diagram to represent all of the steam considered, the area *abcde* represents the energy available from this amount of steam and the equations for the cycle efficiency may be developed readily.

In case superheated steam should be bled, the process cannot be shown in the superheated field by drawing a line parallel to the liquid line. The reason for this is that the specific heat is so much less for superheated steam than for water, that even though all of the throttle steam should be bled there would not be enough heat available from it to heat the feed water to the same temperature as that of the superheated steam at the beginning of bleeding. This statement does not mean that there are no cases in which it is economically desirable to bleed superheated steam; but it is intended to call attention to the fact that the temperature-entropy diagram, the equations, and the efficiency curves given in this paper are to be used only when saturated steam is bled.

It is particularly instructive to observe from Fig. 4 how the temperature of the feed water rises in this cycle as we go to high pressures. For the ideal regenerative cycle using a throttle pressure of 1200 lb. per sq. in. and a temperature of 700 deg. F. the feed water might be returned to the boiler at a temperature of about 450 deg. whereas for a throttle pressure of 200 lb. the feed water temperature wuold be only about 225 deg.

The Reheating-Regenerative Cycle (E) is outlined in Fig. 5. It is called "reheating-regenerative," for the reason that the steam is reheated as in the case of cycle B, and then the regenerative principle is also applied as discussed for cycle D. Therefore the area *abcdegh* represents the available energy in B.t.u. per pound of throttle steam. This cycle implies considerable complication, but under certain conditions its possible advantages apparently justify the building of a plant to operate with this ideal.

The Isothermal-Regenerative Cycle (F) is made up of the processes shown in Fig. 6. This cycle can hardly be called important except in so far as it indicates the limiting possibilities of cycle E with an infinitely large number of reheaters, so that the isothermal expansion in the superheated region is approached so shown by the curve *cd*. The regenerative portion of this cycle is exactly similar to that described for the two previous cycles, except that the saturation temperature at which the bleeding begins is determined by the amount of isothermal expansion permitted in the turbine. In other words, the length of the line *cd* determines the temperature at *e* and consequently at *f*.

Efficiencies of Various Cycles

In Fig. 7 is shown the effect of pressure on the efficiency of the Rankine cycle (A) for the two throttle temperatures 700 and 800 deg. F., both with and without the correction for the ideal feed-pump work.

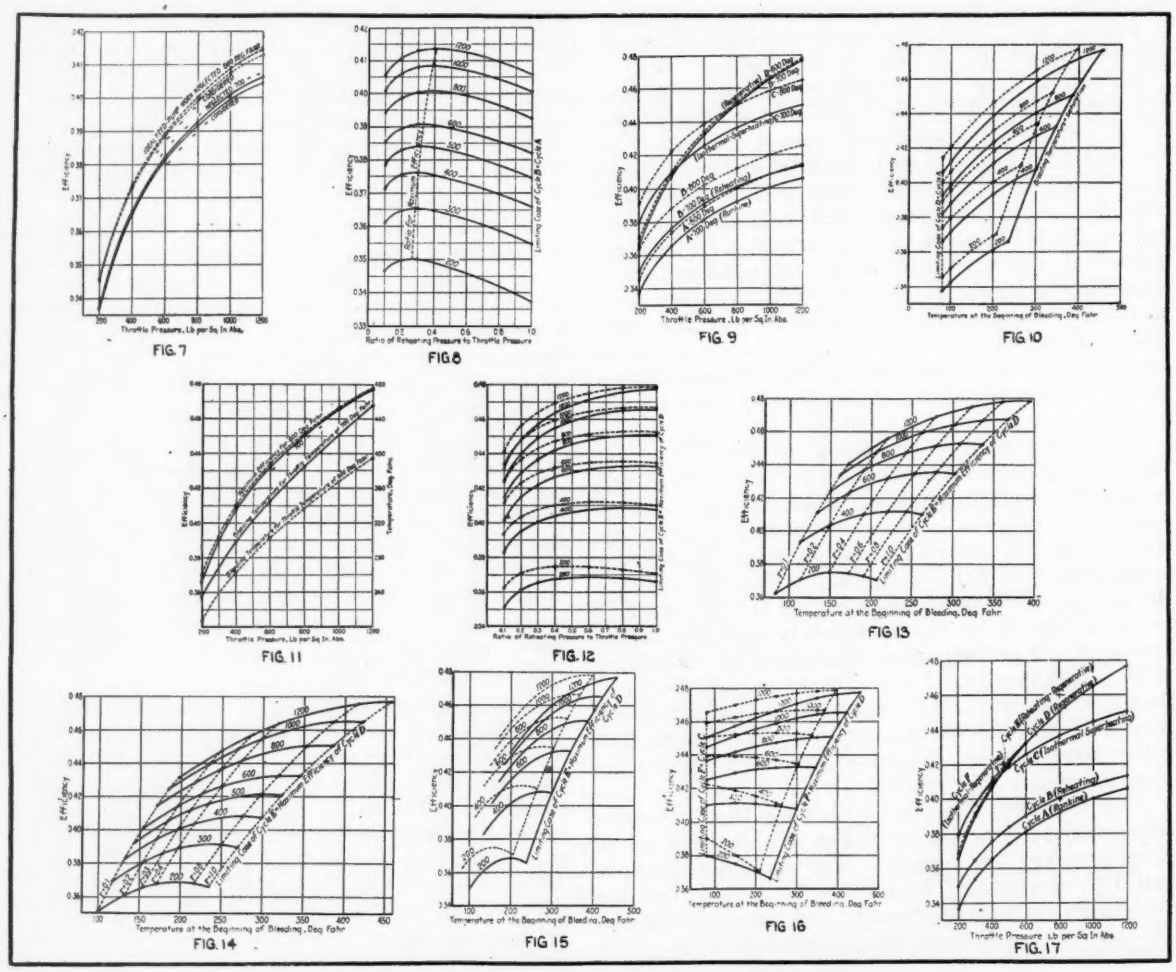
These curves show that increasing the pressure from 200 to 1200 lb. per sq. in. causes the efficiency of the Rankine cycle to increase about 20 per cent, while increasing the throttle temperature from a value of 700 to 800 deg. F. will cause the efficiency to be increased only about 2 per cent.

Efficiencies of the Reheating Cycle (B) for various throttle pressures and reheating pressures are shown by the curves in Fig. 8. The increase in the throttle pressure from 200 to 1200 lb. per sq. in. will cause the efficiency of the cycle to increase by about the same per-

centage as the Rankine, which may be considered as merely a limiting case of cycle B, with the reheating pressure equal to the throttle pressure.

For the Isothermal-Superheating Cycle (C) the curves of efficiency for the two throttle temperatures are shown in Fig. 9. The curves of cycle C for both temperatures are appreciably higher than those for cycle A or B for the same temperatures, a result which is to be expected, since in cycle C a greater portion of the heat supplied during the cycle is added at the maximum temperature.

If there were available no cycles other than A, B, and C, whose relative efficiencies are clearly so favorable to C, one might attempt the difficult task of making a turbine in which the isothermal superheating process is approached, but fortunately there is also the regenerative cycle (D), which has a higher efficiency than C for sufficiently high throttle pressures. Figure 9 shows that cycle D is superior above 450 lb. with an initial temperature of 700 deg. F. and above 750 lb. with an initial temperature of 800 deg. F. Further, cycle D merely calls for bleeder heaters, which are simple and not very costly while C requires a new type of turbine built so as to maintain isothermal expansion.



FIGS. 7 TO 17. EFFICIENCIES OF VARIOUS CYCLES

Fig. 7. Cycle A (Rankine), exhaust pressure 1 in. mercury abs. Fig. 8. Cycle B (reheating), throttle temperature 700 deg. F., exhaust pressure 1 in. mercury abs. Fig. 9. Comparative maximum efficiencies of Cycles A, B, C and D. Fig. 10. Cycle D (regenerative), throttle temperature 700 deg. F. solid lines, 800 deg. F. broken lines, exhaust pressure 1 in. mercury abs. Fig. 11. Maximum efficiency and the corresponding bleeding temperature of Cycle D (regenerative). Fig. 12. Cycle E (reheating-regenerative), temperatures 700 and 800 deg. F. shown solid and broken respectively, exhaust pressure 1 in. mercury. Fig. 13. Cycle E (reheating-regenerative), throttle temperature 700 deg. F., exhaust pressure 1 in. mercury, r = ratio of reheating pressure to throttle pressure. Fig. 14. Cycle E (reheating-regenerative), temperature at throttle, 800 deg. F., exhaust pressure 1 in. mercury, r = ratio of reheating pressure to throttle pressure. Fig. 15. Cycle E (reheating-regenerative), exhaust pressure 1 in. mercury, throttle temperatures 700 and 800 deg. F. shown by solid and broken lines respectively. Fig. 16. Cycle F (isothermal-regenerative), throttle temperatures 700 and 800 deg. F. shown by solid and broken lines respectively, exhaust pressure 1 in. mercury. Fig. 17. Comparative maximum efficiencies of Cycles A, B, C, D, E and F, throttle temperature 700 deg. F. and exhaust pressure 1 in. mercury abs.

Considering the Regenerative Cycle (D) alone, it should be observed from Fig. 10 how the efficiency varies with a change in bleeding temperature (or pressure), throttle pressure, and temperature. For each case it may be seen that the maximum efficiency occurs when the bleeding begins as soon as the saturation line is reached. The reason for the very slight improvement in efficiency resulting from increasing the throttle temperature to 800

deg. F., is that as the throttle temperature is increased the maximum bleeding temperature goes down, as shown by the diagram of Fig. 4. The rising initial temperature tends to increase the cycle efficiency; but the decreasing boiler-feed temperature materially offsets this effect. The relation between throttle pressure and temperature, and the bleeding temperature to give maximum efficiency of this cycle, is shown by the curves in Fig. 11.

Curves which represent the efficiency of the Reheating Regenerative Cycle (E) are shown in Figs. 12, 13, 14 and 15. Figure 12 shows how the efficiency is affected by changing the ratio of heating pressure to throttle pressure, and it also shows that there is a much greater improvement produced by increasing the throttle temperature to 800 deg. F., when this ratio is small. The temperatures of saturated steam at the beginning of bleeding, which change with the various throttle pressures and reheating pressures, are shown by the curves in Figs. 13 and 14 for the throttle temperature of 700 and 800 deg. F., respectively.

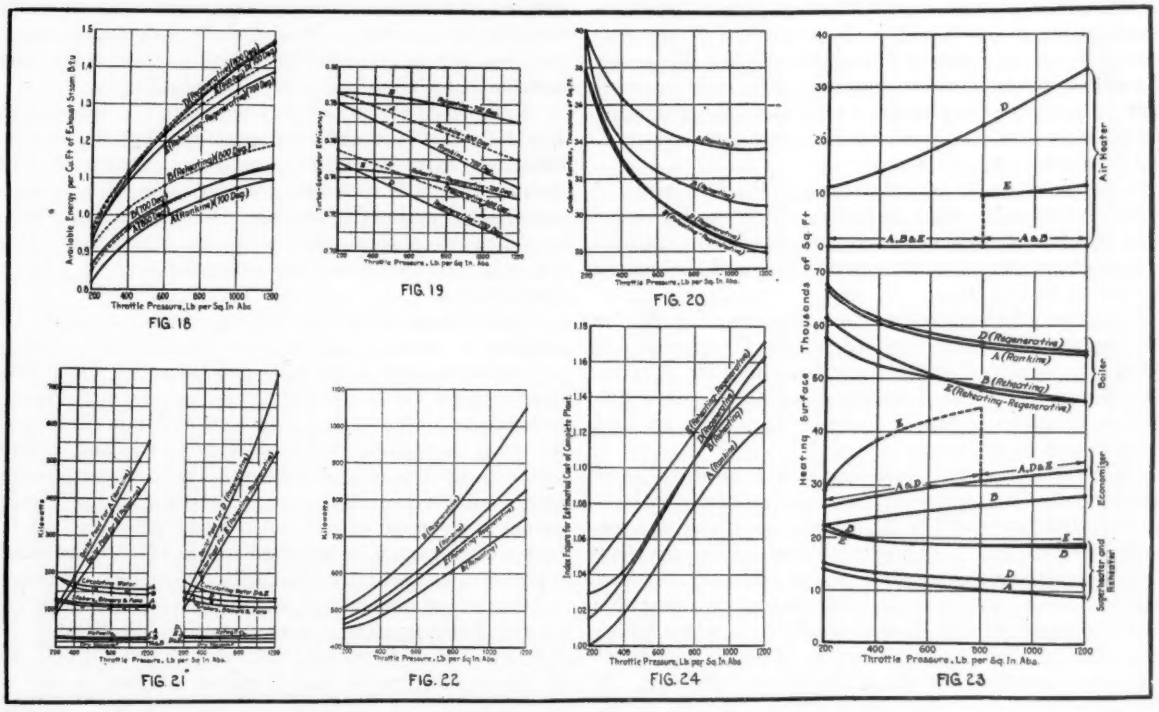
Figure 15 shows the curves of Figs. 13 and 14 combined, except for the omission of the reheating pressure ratio. In all of the curves representing the efficiency of cycle E, it may be noted that only at the lower throttle pressure does this cycle give a higher efficiency than the regenerative cycle (D).

Cycle F (Isothermal-Regenerative) gives efficiencies with values which vary from those of cycle C to those which are equal to the maximum of cycle D, as may be seen from Fig. 16. In other words, the ratio of isothermal expansion in the ideal turbine, V_d/V_e of Fig. 6, has been varied from unity which corresponds to cycle D, to the maximum which corresponds to cycle C. This maximum ratio of expansion is arbitrarily taken such that at the end of the ideal adiabatic expansion the steam is exactly dry and saturated at the exhaust pressure, so that for this case there can be no bleeding.

In Fig. 17 are given the efficiencies of the six cycles arranged so that they may be compared for a throttle temperature of 700 deg. F. and various throttle pressures from 200 to 1200 lb. per sq. in. Clearly, cycle D is the one which deserves first place, especially for the higher pressures, as far as efficiencies on the ideal basis are concerned.

In addition to the efficiency, one other value which is important to engineers and which may be calculated for the ideal cycles is the amount of energy available for work in the ideal turbine per unit volume of the steam at exhaust. This ratio is important because it is a partial measure of the relative sizes and costs of the turbines required to give the same power when operating on the different cycles. The superiority of the regenerative cycle in this regard also, is shown by Fig. 18. Cycles C and F were not considered of sufficient importance to justify drawing their curves in this figure.

For any cycle herein studied, it is clear that there is a marked increase in efficiency and in energy available per unit volume of exhaust steam, due to increasing the throttle pressures, throughout the range considered. This increase in efficiency with the pressure is less



FIGS. 18 TO 24. CURVES USED IN ESTIMATING PLANT PERFORMANCE

Fig. 18. Available energy per cubic foot of exhaust steam, exhaust pressure 1 in. mercury. Fig. 19. Probable efficiencies of 30,000-kw. turbo-generator. Fig. 20. Estimated condenser surface for 30,000-kw. turbine, throttle temperature 700 deg. F. Fig. 21. Estimated auxiliary power for boiler and condenser rooms. Fig. 22. Estimated total auxiliary power required for a gross turbo-generator output of 30,000 kw. Fig. 23. Heating surface required to give an efficiency of 84 per cent. Fig. 24. Estimated relative costs of complete central stations, including buildings and land.

marked in the simple Rankine cycle than in the ones with more complicated apparatus.

Changing the throttle temperature from 700 to 800 deg. F. causes a marked increase in the efficiency of all cycles herein studied except those using the regenerative principle, which are improved but little by this increased temperature.

Probable Results from Real Plants

The efficiency of each of the ideal cycles may be calculated so that the results are correct except as they may be slightly affected by uncertainty in the values of the properties of high-pressure steam. The real steam plant, however, can never do as well as its corresponding ideal, and the degree of approach made toward this ideal is a very proper measure of excellence of the design, construction, and operation of such a plant.

Of all the power-plant apparatus involved in a cycle the turbine is the one whose efficiency is more sensitive to changes in the cycle and in the pressure and temperature of the steam. Instead of dealing with the efficiencies of turbine and generator separately it is more convenient to work with the combined unit, since the generator efficiency remains constant for the same speed, output, and power factor.

Efficiencies to be expected from turbo-generators of 30,000 kw. capacity when operating on the cycles under consideration with different pressures and temperatures have been calculated as well as possible, and the results are shown in Fig. 19.

The amount of cooling surface required, Fig. 20, for the condenser was based on a heat-transfer rate of 470 B.t.u. per sq. ft. of surface per hr. per deg. F. mean temperature difference. A fair mean temperature difference between the water and the steam was believed to be 15 deg. F.

Calculations of the auxiliary power required were made for the four cycles and the four pressures as indicated by the curves of Fig. 21, which shows the power required by each auxiliary motor. These curves also show very clearly how the power required by the boiler-feed pumps rises as the pressures is increased and how this power goes up with the increased flow of water which is necessary with the regenerative cycles D and E, as compared with those in which there is no bleeding. In Fig. 22 may be seen the curves which show the power required by all the auxiliaries for each of the four cycles and all pressures from 200 to 1200 lb. per sq. in.

Relative amounts of heating surface required for the boiler, economizer, superheater, reheater, and air heater are perhaps best indicated by the curves of Fig. 23. It will be noted that for the high pressures, both of the regenerative cycles require an air preheater in order to give an efficiency of 84 per cent. This is due, of course, to the high feed-water temperature. For pressures less than 800 lb. per sq. in. the use of preheaters is not so essential with the reheating-regenerative cycle E as it is with D, which has no reheating.

Initial Costs of Plants

If to the cost of equipment affected by steam pressure there is added the cost of land, canals, railroads, foundations, buildings, coal handling, miscellaneous mechanical equipment, and all electrical equipment except the generator, index figures are obtained as shown by Fig. 24. From these curves it would appear that the 1200-lb. plant will cost only about 14 per cent more than the 200-lb. one to operate on the same cycle. For many cases this increase in cost will be greater when the higher pressures are used, for the reason that some plants will have a higher portion of the total plant cost dependent upon pressure than was used in this study. It is also clear that the plant built to operate upon the Rankine cycle is appreciably lower in first cost than any of the others, and the reheating-regenerative plant is the most expensive throughout the entire range of pressures considered.

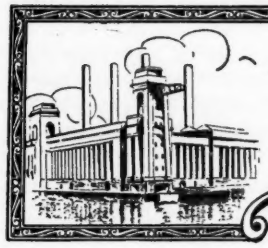
Experience with high-pressure equipment and with auxiliary apparatus which the use of high-pressure steam appears to involve is as yet exceedingly limited. Nobody is today really in position to say positively that the high-pressure equipment now coming into use will operate with that smoothness and certainty required to justify its use for the sake of the increased thermal efficiency obtainable. Under such conditions, it would seem to be best to use first the simplest types and to progress toward the more complicated if further experience indicates such progress to be desirable.

From such a point of view the simple Rankine cycle would be best if it gave sufficient promise of increased efficiency with increased pressure. Unfortunately it does not, and it is therefore necessary to consider a more complicated cycle of operation. Our present knowledge indicates two choices, that involving regenerative heating only and that involving reheating with or without regeneration.

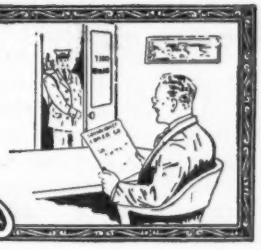
Considering all of the factors which enter into the problem, it seems to the authors that the high-pressure regenerative plant is the most promising for commercial development. It is certainly true that the performance here estimated for such a plant can be further improved by using a turbine designed to separate and remove water formed during expansion. Some of the designs now available provide for such drainage at the bleeder points to a certain extent, but it seems probable that the best results cannot be attained unless effective moisture separators are actually built into the turbine structure.

Certain of the factors in this paper have been calculated for maximum temperatures of both 700 and 800 deg. F. The authors feel that calculations for 800 deg. F. are at the present time of minor interest, as they do not believe that materials and designs now available may be considered adequately safe for use with such temperatures. It is exceedingly difficult to state categorically any definite upper limit of temperature, but it is felt that a figure between 700 and 750 deg. F. represents the safest upper limit of steam temperature at the present time.

The authors believe that this study indicates that even with the present type of turbine the regenerative cycle is the best one to use in large stations, as it ranks very high from the standpoint of fuel consumption, operating characteristics, and first cost throughout the entire range of pressures, but particularly so above 600 lb. per sq. in. With a turbine which the authors believe can be developed so as to remove the moisture to a considerable degree, this cycle will give still better economy, and it seems altogether reasonable to expect that this equipment in the turbine room, combined with that now developed to yield high boiler-room efficiency, will give station economy that will pay handsomely for the increased investment.



LETTERS DIRECT FROM THE PLANT



Insulation of Furnace Walls

COMMENTS BY C. O. Sandstrom are presented in the Aug. 1 issue of Power Plant Engineering on page 794, on my article which appeared in the June 15 issue.

Mr. Sandstrom seemingly has not had the opportunity to read Mr. Edwin Ricketts' paper on boiler furnace design, which appeared in the May 1 issue of Power Plant Engineering. Inasmuch as my paper was a comment on that presented by Mr. Ricketts, the figures used were taken from Mr. Ricketts' paper.

It is a well recognized fact that the furnace wall temperature is difficult to determine. For this reason, no doubt, Mr. Ricketts assumed 2500 deg. F. on the furnace wall. This was assumed directly on the brick and consequently the emissivity factor was eliminated.

As brought out in my previous comment, the temperature on the furnace wall is dependent to a great extent on the pressure conditions obtaining in the boiler furnace. Mr. Sandstrom seems to have overlooked this important point. Tests have been conducted which show the temperatures in furnace walls. Mr. Sandstrom makes the point that all figures on furnace wall temperatures are based on theory. Those interested are referred to an article by Mr. Joseph Harrington in the March 27, 1917, issue of Power. The tests conducted by Mr. Harrington show the temperatures existing in the different sections of the walls, as measured by pyrometers.

It is not particularly difficult to heat an iron rod to red heat when the rod is imbedded in a wall and is entirely surrounded by brickwork. The steel shell comparison cited by Mr. Sandstrom is hardly analogous. Furthermore, the operation of a cement kiln can hardly be compared to boiler operation. Temperatures in excess of 3000 deg. are seldom obtained in a cement kiln, and it may be safely stated that the maximum temperature reached seldom exceeds 2700 deg. F. This temperature obtains only in the burning or clinkering zone of the kiln. The steel shell in this section of the kiln is usually protected by 9 in. of firebrick and 6 or 8 in. of cement clinker which has stuck to the firebrick. From time to time this clinker drops off.

In the cement kiln, the steel shell is exposed and is constantly revolving. This necessarily increases convection losses, which together with radiation, allows considerable heat to be dissipated. Mr. Sandstrom's frequent references to cement plants lead me to assume that he is familiar with conditions in such plants. It is suggested that he determine the temperature on such shells by actual test.

Mr. Sandstrom states that the "I" beam supporting two 400-hp. boilers, which he recently dismantled, was not injured by the heat, even though it was right next to the furnace section. It will be of particular interest to note that all public utility plants are protecting such supporting beams with proper insulation materials. The elastic limit of steel does not depend on the temperature by definition. It is described by Kent as that load at which the deformation ceases to be proportional to the stresses or at which the rate of stretch begins to increase.

While it is possible to purchase 3000 or 400 red brick for approximately the same price as 1000 insulating brick, the labor cost for laying the red brick changes the situation entirely. It is customary to assume that red brick can be laid up for $45 or $50 a thousand, including material and labor; 1000 insulating brick can be purchased and installed for approximately $115 to $125.

If Mr. Sandstrom's statements regrading air infiltration are based on tests, his information along this line would be quite valuable to other engineers. His criticism of the writer's article refers particularly to the assumption made, which seems to indicate that his statement that insulating brick increase air infiltration is based on theory.

Small scale tests will show that red brick are thermally much more conductive than insulating brick. It is a well recognized fact that the conductivity of red brick is approximately eight times that of a high grade insulating brick.

In the June 1 issue of Power Plant Engineering, page 573, there is a description of the new Marysville plant of the Detroit Edison Co. The furnaces, on these boilers, which each have an effective heating surface of 28,212 sq. ft. are 18 ft. 9 in. high, 26 ft. 7 in. wide and 19 ft. 4½ in. long, which gives a wall area of the furnace of approximately 1160 sq. ft. I know that Mr. Sandstrom will be interested in this article, inasmuch as he feels that the size of the furnace assumed in my comments on Mr. Ricketts' paper was entirely out of line. It really makes little difference what the size of the furnace is, so long as the temperature exists. It requires only a certain number of firebrick and a certain number of insulating brick, and a certain number of red brick to insulate one square foot of surface, regardless of whether that surface is on a small or large boiler.

Savings made possible by the use of insulation in boiler walls warrant the earnest consideration of every engineer who has anything to do with boiler construction and operation. C. A. FRANKENHOFF.

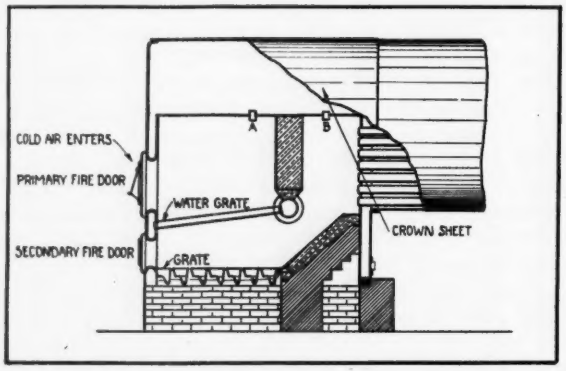
Fusible Plugs

DURING THE course of my work as boiler inspector, I have found that without exception the fusible plugs of a certain type of down draft heating boiler were in a position which I believe renders them to be of little or no value.

Apparently it is the practice of the boiler builder to place the fusible plug in the position shown at "A"

in the accompanying sketch, which is above the primary fire grate. This is in accordance with a literal interpretation of various boiler codes, which usually state that fusible plugs should be installed in the highest point of the crown sheet of locomotive type boilers. However, mere compliance with the letter of codes is not the end in view; protection against low water is desired. A temperature of 400 to 500 deg. F. is necessary to melt the standard pure tin plug, and while there may be times when the space above the primary fire grate attains a temperature of 400 to 500 deg. F., this is not true all the time when the boiler is in operation because when down draft boilers are working up to capacity, it is necessary to admit cold air above the primary grates. This condition lowers the temperature considerably and I am sure renders the fusible plug ineffective.

Position "B" shows the plug installed in that portion of the crown sheet which is at all times exposed to



TYPICAL DOWN DRAFT HEATING FURNACE SHOWING USUAL LOCATION OF FUSIBLE PLUG AT "A"

the products of combustion. I believe this is the proper place, and I always recommend that the fusible plug be installed in this position.

It would be interesting to hear comments from other inspectors or engineers on this point.

Oklahoma City, Okla. WALTER H. FITTUS.

Small Pulleys

WHENEVER I see the claim made that manufacturers often put small pulleys on machines just to make the buyer believe that the machine runs light—that little power is required to drive the machine—I am inclined to believe that there is a grain of truth in the claim. A big pulley always looks rather formidable and hard to drive, whereas a smaller pulley has a more innocent look. I have studied belts and pulleys to a considerable extent and concluded some time ago that large pulleys are best. They make possible a narrower and less expensive belt; they are not so liable to slip; they will transmit more power; the belt tensions need not be so high to pull the load; the belt will therefore last longer; there will be less power loss in the journals; and often the total cost of the two large pulleys plus the belt is less than would be the total cost of two small pulleys and a wide belt.

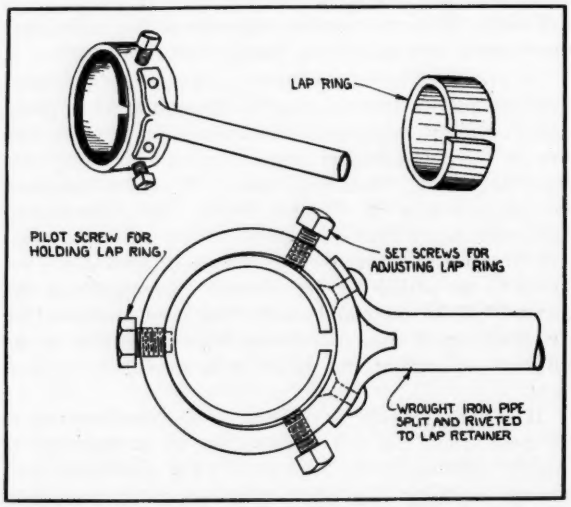
Again, I believe that designers put small pulleys on small machines because they just make a "guess" as to the proper size. They don't think that a high speed of, say, 4000 ft. per min. is more economical than a speed of half that much. On a machine 10 ft. high they put, say, a 2-ft. pulley. On a machine 5 ft. high, they therefore figure that a 1-ft. pulley is sufficient, evidently ignoring the fact that the belt speed will be cut in two. They follow out about the same proportion in the design of all machines—large pulleys on large machines, and small pulleys on small machines. The manufacturer likes to pretend that the power required to drive his machine is almost nil. This is obviously wrong.

N. G. NEAR.

Renewable Lap for External Work

IN THE accompanying sketch, a special form of external lap is shown, which was used advantageously for finishing the surfaces of piston rods. The requirement of a polished uniform surface for the rods to work smoothly in the packing glands without rapidly cutting out the packing justifies the use of lapping to obtain the finished surface.

In the construction shown, the lap surface, being subject to considerable wear from the quantity of work,


LAP RING
PILOT SCREW FOR HOLDING LAP RING
SET SCREWS FOR ADJUSTING LAP RING
WROUGHT IRON PIPE SPLIT AND RIVETED TO LAP RETAINER

LAPPING RINGS MADE FROM SECTIONS OF BRASS PIPE ARE RENEWABLE

was made as an adjustable and renewable section of the lap.

These parts were made from brass pipe, turned to size and split. The handle section was made in the form of a retaining ring, with securing and adjusting screws. The handle was made 12 in. long, to facilitate holding the lap steadily at the work. The retainer is a forged steel ring, with a handle made from a length of pipe, split at one end and riveted to the ring. The brass laps were turned out in quantity when the machine was set, to machine these up. New laps are then available when required to replace one that is worn out. In use these laps are charged with emery and oil, the shafts polished until every evidence of tool marks is removed.

The one lap handle can be used if desired, for various diameters of laps, a desirable condition, that can be obtained in an inexpensive construction.

Washington, D. C. G. A. LUERS.

Change of Brushes Eliminates Sparking

RECENTLY, WHILE looking over my Sept. 1 issue of Power Plant Engineering, I read A. R. Knapp's "Pointers on the Maintenance of Commutators" for the second time; for there I found considerable good information relative to the general care of motors.

In one paragraph, he takes exception to the changing of brushes as a remedy for some commutator troubles. This, to my mind, is a mistake; for in two cases that have come to my attention that was the only apparent solution for certain poor operating conditions.

Several years ago, a new 220-v. direct current motor, belted to a bucket coal conveyor which came under my care, for a long time gave trouble at the commutator. There were four sets of brushes: two brushes to each holder. The top and bottom set would wear out in two weeks and the commutator would score badly in about one month. The two side sets of brushes would be in fairly good condition. There was no sparking at the brushes, the motor ran well and everything which we thought could possibly be the trouble was eliminated, but without results. Finally the chief engineer brought in a set of new brushes of a different make from those in use and asked me to try them out.

The armature was removed and placed in lathe and the commutator turned and placed in first class condition and returned to the motor. The new brushes were fitted to the commutator and the motor started.

For the first three days, there was a loud squeaking from the brushes and wiping the commutator with an oily cloth did no good. On the fourth day, however, the noise grew less and in about one week I never saw a nicer commutator.

I remained 3 yr. longer in that power plant and during that time neither the brushes nor the commutator gave us a minute's trouble and when I left both looked good for another 3 yr. This was one case where changing the brushes seemed to be the only remedy.

In this same power house there were two 400-kw. d.c. generators which were new when I took charge. From the start there was more or less trouble from sparking and noise. On one machine, the brush holders proved to be out of line. Straightening them helped some but did not eliminate the trouble entirely.

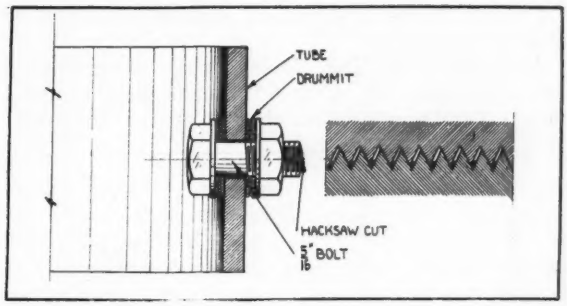
A set of imported brushes were purchased for each machine and these were found so soft that we expected they too would be a failure; however, they eliminated all of our brush and commutator troubles and after 4 yr. in service the wear on commutator and brushes was not perceptable.

New York, N. Y. A. K. VRADENBURGH.

Temporary Repair for Leaky Boiler Tube

ONCE UPON a time, away back in 1909, at our plant on the water front at Long Beach, L. I., the sea water got into the fresh water supply line and before I realized it I had 19 grains of salt per gallon in the boiler water. About 8 o'clock, the night engineer reported to me a tube leaking in the 150 hp. Stirling boiler. No tubes were on hand in the plant and I needed this unit to carry the day load. Remembering an old kink that seemed to apply here, I took a 5/16-in. bolt, ¾ in. long and made a hacksaw cut through the center of the end. I then made a drummit (gasket) by interweaving stranded copper wire so that it fitted the bolt, and smeared it over with some white lead. I then got a single strand from a No. 14 lamp cord and wound it into the threads tying the knot in the hacksaw cut at the end of the bolt.

To apply this bolt I placed two planks over the grates, and went in the fire box with a flat file and with the butt end, made a hole large enough to receive the bolt. In order to get the bolt in the tube, I placed a man in the mud drum and told him to hitch the end of the bolt to a piece of bale wire and work it down the tube. When the bolt arrived opposite the hole I put a washer on the outside and tightened up the nut.

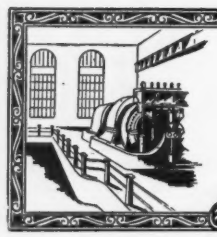


BOLT WAS PUT THROUGH THE BURNT OUT HOLE IN THE TUBE. MATS OF COPPER WIRE SERVED AS GASKETS

This arrangement held for 10 days until I was able to procure a new tube to make a replacement. POP.

Live Steam Cools Hot Turbine

IN THE Nov. 15 issue, on page 1154, John U. Morris tells of an interesting installation that he came across in visiting a generating station in Tacoma, Wash., where a 6000-kw. turbine unit was used as a synchronous condenser during the summer months, to increase the power factor on a line supplied largely by waterpower. He speaks of the turbine being run idle but wishes to know why a certain amount of live steam was used to cool it. While it does seem rather strange to use live steam for cooling, it nevertheless is the practical thing to do in this case. It is not unusual to notice discoloring of the turbine blades from overheating when air enters the machine. The speed of such bucket wheels is so high that if the machine is full of air the windage friction in churning the air around through the wheels is great enough to generate considerable heat or even to destroy the buckets. Keeping the machine full of steam allows the wheels to revolve in a lighter medium, especially if an exhaust pump is used. Less power is, therefore, required, while the moisture in the steam will cool excessive hot parts to a safe temperature. I came across some of these experiences while testing steam turbines in the General Electric Co. some years ago. As to why it should require much more steam during the noon day than during the night is not clear to me, but I think if a careful investigation were made it would be found that other causes such as leakage may have a greater effect on the amount of steam used than that caused by the difference in temperature of the room, although variations in the amount of moisture in the atmosphere may affect the seal on the carbon packing ring. P. G. B.



PROBLEMS IN OPERATING POWER EQUIPMENT

How Much Heating Surface Will Be Required?

I WOULD greatly appreciate a solution to the following heating problem:

A one-story building, similar to the accompanying sketch, will have 58—4 by 6-ft. single glass windows spaced around it, approximately 8 ft. apart. The height

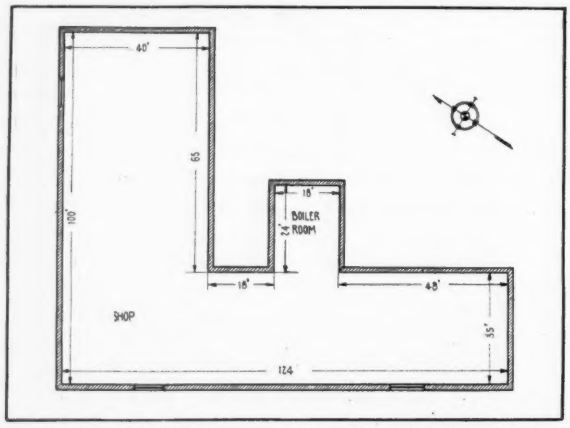


DIAGRAM OF BUILDING TO BE HEATED, INDICATING THE EXTENT OF EXPOSED SURFACES

of roof finished, which will be of tar and gravel construction, will be 12 ft. Walls of the building are to be concrete blocks 12 in. thick. There will be 16—14 by 20 by 2-in. heaters for process work. They will be of cast-iron construction and on an independent steam supply from the boiler.

How many lineal and square feet of 1¼-in. pipe would be called for?

What size steam main must I use?

Using bituminous coal as fuel, what size horizontal return tubular boiler must I use and what should be the size of the stack?

In an installation of this kind, would the size of the plant warrant the use of an oil burning system?

The boiler pressure required will be 15 lb. per sq. in. Would it be advisable to put in a reducing valve and run the heating system under a vacuum?

Would this be necessary with such long runs of piping, and with such a small drop of the return lines?

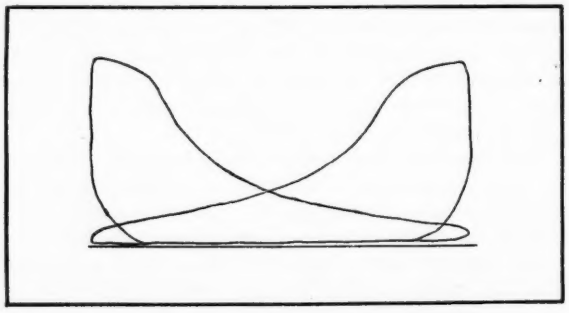
L. B.

Would Additional Radiant Heating Surface Be Useful?

WE CONTEMPLATE rebaffling one of our boilers and would like to know what advantage, if any, could be gained by covering the second row of tubes, instead of the last, in a horizontal baffled, horizontal water tube boiler, burning anthracite coal. C. P. W.

What Causes the Drop in the Steam Line?

I WOULD like to have an opinion from your readers as to what is the matter with the valve setting on the engine from which the accompanying indicator card was taken. What causes the slanting steam line from



DOES A RESTRICTED ENGINE LEAD CAUSE THIS DROOPING STEAM LINE?

the point of admission? The engine is a 16 by 36-in. double eccentric Reynolds Corliss running at 120 r.p.m. This engine is about 20 yr. old.

The steam header is 8 in. in diameter and engine connection is 5 in. and about 18 ft. long. It would appear from the indicator that there is a loss of pressure between the header and the engine; but this does not seem possible. Is it caused by the way in which the valves are set?

The card was taken with 120 lb. steam pressure and 108 kv.a. load. The engine is direct connected to a 200-kv.a., 2300-v. alternator. L. F. W.

Governor Drive on Corliss Engines

IS IT true that all Corliss engine governors of the centrifugal flyball type are driven by belt from the engine shaft? If so, what is the reason for it? It seems that a belt drive for this extremely important part of an installation is rather uncertain and means taking unnecessary chances. Could not the governor be operated by a light lay shaft and gears instead of by belt?

J. B.

Utilization of Exhaust Steam

IN THE Nov. 1 number, E. W. M. mentions some difficulties that he is having regarding the use of exhaust steam for heating. First, he speaks of heating an office building some 250 ft. away from the plant. It is assumed that he has adequately insulated supply pipes of ample

size to supply the radiation in the office building. The difficulty then seems to be that of raising the returns 3 ft. and pulling them back to the engine room with a vacuum pump located in the engine room basement. He does not state whether the engine room floor is 3 ft. higher than the floor of the office basement; but in many cases of this kind, the radiation is hung from the ceiling of the basement so that there will be no difficulty about the returns.

Even if the return lines are near the basement floor, it would seem that 15 in. of vacuum would be sufficient to pull them back. There are lift fittings that may be used in some cases to aid this flow. Another method would be to place the return pipe in a trench 3 ft. deep with the vacuum pump in a pit in the engine room. A vacuum pump could also be placed in the basement of the office and if current is available at all times, an electrically driven pump could be employed. This would avoid the necessity of running high pressure steam to the office building. The discharge of the vacuum pump should be piped to the open heater so that it can be used for feed to the boilers.

There is no reason why the coils in the dry rooms should not be supplied with exhaust steam in summer, or at any time, when it is available. The connections to the high pressure traps should be changed and low pressure vacuum traps connected and the returns piped to the suction line of the vacuum pump. Often, after this is done, there is nothing else to do but to connect the supply to the exhaust line and the coils will heat as before. Sometimes the supply line is too small or needs insulating. It might be necessary to add some extra radiation to compensate for the lower temperature of the exhaust steam.

There should be no necessity for using a 10-lb. back pressure if the piping is of the right size, as this causes the engine to take much more steam than is necessary to do the work.

East Dedham, Mass. G. H. KIMBALL.

Oil in the Feed Water Heater

IN REPLY to F. G. F.'s question in the December 1 issue, on page 1206, concerning oil in the feed water heater and cylinder lubrication, I offer the following.

It seems clear that more than sufficient oil has been fed to the engine cylinder, even though in the opinion of the operators everything appears to be correct in that respect. Operators are prone to overfeed cylinder oil to engines, and when trouble with oil coated surfaces is experienced, they usually work on the effect and not the cause.

It does not require as much oil to maintain a lubricating film between rubbing surfaces as is usually thought necessary. Especially is this true if the correct oil for the job has been chosen from an analysis of the factors involved, such as size, load, quality of the steam, pressure, and speed. The relation that these factors bear to the lubrication of steam cylinders has been treated before in these pages, and I mention it now for reference only. For the present problem, a heavy body, compounded, high grade cylinder oil is required to meet the high pressure (comparatively so called) and wet steam conditions. A high grade compounded oil will resist the working off effect and will "stay put" as we say, or, it has greater "stiction," to use a coined word. On the contrary, if a straight mineral cylinder oil is used where the steam is wet, about twice as much of it will have to be used to accomplish the double purpose of providing adequate lubrication and effecting a piston seal.

Because the straight mineral oil will not emulsify with the water of condensation in the engine cylinder, much of it will go out with the exhaust steam and deposit where it may.

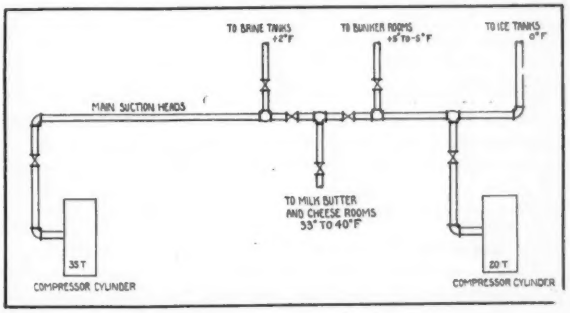
Let F. G. F. try out this procedure. First, make sure that the oil is compounded and of good grade. Reduce the feed to about from 6 to 8 drops per minute for each cylinder and watch the lubricator for uniform feeding, if it is of the hydrostatic type. Install an atomizer in the steam pipe for good distribution of oil. At the end of a week, or any convenient time when the engine is shut down, take off the cylinder head and examine the surfaces for the quantity of oil remaining on them. It can easily be seen whether or not too much oil has been fed. The oil feed should be reduced to such an extent that upon periodic inspections, the surfaces show that they are not absolutely dry.

Under these conditions, I am of the opinion that with the quantity of oil suggested, little if any oil will go out with the exhaust, and that is the primary object sought. Most operators are so afraid that the interior parts of the engine will run dry, they overflood them with oil, in order to be on the safe side, as they think. The object should be to run with as little oil in the cylinders as possible, consistent with the necessary lubricating and sealing film. More than enough for these purposes must result in the surplus going out in the exhaust steam.

Brooklyn, N. Y. CHARLES J. MASON.

Change in Ammonia Suction Piping

ON PAGE 1206 of the Dec. 1 issue, B. M. asks how to arrange his ammonia suction piping to his two compressors so that he can operate either unit on any part of his load. It is desirable, he says, to use the larger unit for the lower temperature work and the smaller machine on the 40-deg. rooms. The accompanying sketch will serve to show how this arrangement may be carried out. All leads except the one to the ice tanks are connected into a common header between the two units.



ARRANGEMENT OF AMMONIA SUCTION PIPING THAT WILL ALLOW EITHER COMPRESSOR TO CARRY ANY PART OR ALL OF THE LOAD

Rochester, N. Y. R. G. SUMMERS.

POWER PLANT ENGINEERING

Trade Mark, Reg. U. S. Pat. Office

Published by the Technical Publishing Co.
537 South Dearborn Street, Chicago.

Entered as second class matter May 1, 1908, at the Post Office at Chicago, Illinois, under the act of March 3, 1879. Issued the 1st and 15th of each month. Established over 26 yr.

Subscription price $2.00 a year in the United States and all its possessions. Canada $2.75 a year. All foreign countries $3.50 a year.

Copies of current issue, 15 cents; of back issues, 25 cents.

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Member, Associated Business Papers.
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New York Office, 150 Nassau St.

Circulation of this issue, 21,000.

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Utilizing Dual Value of Steam

To utilize every heat unit that the fuel burned contains, is the ideal toward which the power plant engineer strives. The degree to which this ideal may be attained depends upon many factors surrounding the plant. These factors are not all under the control of the engineer, yet in many cases the engineer accepts the conditions as he finds them without any effort to bring about more economical load conditions.

In the industrial plant particularly, there are many opportunities for saving heat that are passed up largely because the extent of the possible savings is not appreciated by the engineer or the owner. Usually the owner is contented when he does not see any exhaust steam issuing from the roof of his power house. He cannot see the amount of live steam that is passing through reducing valves for use in heating the factory or industrial processes, nor does he appreciate the value of the expansive power of this steam if properly used in an engine or turbine before passing to the heating system.

To obtain the highest value from steam under pressure, not only its expansive power but its remaining heat above that of the atmosphere must be used and to secure a load balance between the demand for power and that for heat is one of the most difficult problems which the engineer has to face. Obviously no two plants are alike but the method of solving the problem in one plant as related by Mr. Gould on another page may suggest a means of securing a heat and power load balance in other plants. The savings which can be realized in most cases warrant a careful survey of conditions which affect economy and are subject to modification.

Selection of Motors for Pumps

Because of the wide application of electrically driven pumps in modern steam power plants, it is essential that the engineer be familiar with, not only the characteristics of various types of pumps but also with the characteristics of the motors that drive them. Motor driven pumps are operating successfully under practically all classes of service in industrial plants, but unless care be exercised in the selection of both pump and motor for a particular purpose, the best results will not be obtained. The selection of pumps, motors and control, is not a simple problem, and if not well studied out, the plant is penalized continuously throughout the years the pump is in service.

In the pages of this issue, R. H. Rogers presents an article in which he discusses the application of electric drive to pumping operations. This article should be of considerable interest to power plant men, as it is written from a practical viewpoint, and discusses the relative advantages of various motors for various services.

Honest Buying

There is often a tendency among those responsible for the purchase of mechanical equipment to practice buying methods which will not stand a close scrutiny from all angles. The situation is often taken advantage of just as the borrower of an automobile so often fails to operate and care for property properly which is not his.

Purchasers who buy on price only will fail to obtain quality. The practice of disclosing competitive bids

may lead to a slight saving on the initial cost but repair parts are likely to cost such a purchaser more than if his initial dealings had been absolutely above board. A manufacturer is in business to earn a fair return upon his investment. Is it not reasonable, then, that "gifts" and entertainment must be charged along with production costs, but differ from them in that they represent no value whatever in the goods purchased?

After a manufacturer has repeatedly bumped up against a buyer who insists upon having the price cut, that manufacturer will learn to add the cut to his original price, and the buyer will, in the end, pay a fair price and the one which would have been submitted in the first place had the buyer been reasonable.

Equipment should be purchased on the unbiased judgment of adaptability, quality, serviceability, and price. It should be purchased without selfish motives on the part of the buyer, and any and all advantages which may come from purchasing should revert in full to the company whose money is being expended.

Make your recommendations and purchases as if you were spending your own money for equipment for which you alone will be absolutely responsible.

Off Duty

"Off Duty!" exclaimed the Snap Dragon in disgust, "there ain't no such animal."

"You said it, dearie," sighed the Mignonette, wearily, drooping her pretty head. "I'll tell you frankly as one flower to another, I haven't had a decent night's rest since they put us on this overtime shift. It's not right—"

"Well, what are you two dames crabbing about now?" interrupted the Lima Bean, from the other side of the green house. "Here, they have gone to the trouble of installing a wonderful set of electric lamps to make you grow at night, and now you are kicking about it. At any rate, it doesn't seem to be harming you, you are both looking healthier than ever. Confound it, if any plant around here has cause for complaint, I have. Here I'm stuck away under a dark cover while you two are basking in glorious rays of electric light."

"Oh, Bean, will you keep still? Just because they put you under cover at night you imagine you are being slighted. Even if we do look healthier, can't you understand that we are losing our youthful grace and charm, and that we will be old before you even begin to grow a crop of beans?"

We did not wait to see how the foregoing argument ended, but we don't doubt that the air was full of flowery language while it lasted.

We don't blame Miss Snap Dragon, however. She has just cause for complaint. Eight or ten hours a day work is enough for any self respecting plant, and when we attempt to make her grow 18 or 20 hr. a day, it is overdoing a good thing. From a scientific standpoint, it is of considerable interest to know that the growth of plants can be accelerated by the action of artificial light after sunset, but looking at it from the plants' point of view it is all wrong.

In spite of all opinions, pro and con, the plants thrive on it, however, and in a six weeks' test recently completed by the Westinghouse Lamp Co., many of the specimens exposed to the action of electric light, grew to approximately twice the size of similar plants receiving daylight illumination only. They were considered by experts to be from 14 to 27 days in advance of normal growth.

On the first of September, 12 varieties of vegetables and 12 of flowers were sown in shallow boxes. The soil used, was ordinary light sandy soil such as is generally used for starting seedlings. Two sets of boxes were sown, one set for growing under daylight with artificial light at night, and the other for growing with daylight only.

Both sets were grown under identical conditions of heat and moisture on opposite benches. When electric lights were turned on, an oil cloth curtain divided one group from another. The light was turned on at 8 p. m. and was switched off automatically at 1 a. m.

Almost from the outset there was a marked difference noted in the germination of the seedlings under the electric light. Germination was hastened in some cases to the extent of 4 to 12 days. The advantage gained with advanced germination was maintained in every case through to the end of the experiment. The foliage presented a richer, greener and more vigorous appearance than that of the plants that were denied the advantage of additional light. Also in the case of the vine crops, cucumbers, and muskmelon, the stems of the plants were heavier and stockier than those grown without electric light.

This is all extremely interesting and we have no doubt but that the future greenhouse will be fitted out with banks of powerful lights similar to motion picture studios.

Gosh! what's the world coming to, anyhow? Imagine a florist receiving a rush order for a bunch of orchids at 5 p. m. on a dark, rainy day, and the only ones he has, haven't even begun to bud.

What does he do? Is he dismayed? Does he dance and rave around and call upon the gods for sunlight? Does he? Indeed no, he does not. Instead, he merely goes over to his switchboard, pulls a remote control switch and Presto! his orchid beds in a distant green house are ablaze with light. The next morning he gathers his orchids.

However, laying all joking aside, there is something of real value in this development, and it should promise great things to the horticulturist. Just exactly what it will mean is difficult to predict. It seems, however, to promise a means of doubling the speed of development of many species of plants, and bids fair to enable the horticulturist to bring his flowers into bloom at the time when their market is at its best.

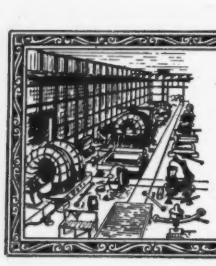
At any rate, if this thing is placed upon a commercial footing, it is certain that the florist will have to add another species of plants to his collection.

"What kind of a plant?" do we hear you ask.

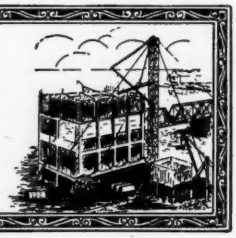
Why, an electric plant, of course, to furnish the light.

Recognition of coal as the basic fuel, and the conspicuous part it has always played in industrial activity, has made it worthy of the intensive study of men representing the best of engineering talent.

Before exact methods of burning coal could be developed, a thorough knowledge was needed of the innumerable factors affecting combustion. Consequently during recent years the science of burning coal has developed into one of the most intricate and highly specialized branches of engineering.



POWER PLANT ENGINEERING DEVELOPMENTS AND NEWS



Mechanical Engineers Discuss Power Generation

AT THE 44TH ANNUAL MEETING OF THE A. S. M. E. COAL STORAGE, HIGH-PRESSURE STEAM, REHEATING AND STAGE FEED-WATER HEATING PROBLEMS BRING OUT SOLUTIONS POINTING TOWARD FUTURE PRACTICE

MANY PAPERS of interest to power plant engineers were presented and discussed at the forty-fourth annual meeting of the American Society of Mechanical Engineers held in New York City on Dec. 3 to 6. Although from Tuesday morning until Thursday evening the problems uppermost in the minds of engineers responsible for power generation seemed to fill the program, the sessions of particular interest were those held Wednesday and Thursday mornings devoted to the subjects of coal storage and steam power plants, respectively.

In his presidential address, John L. Harrington, retiring president of the society, spoke of the position of the engineer in industry as one pre-eminently of fact-finding and analyzing. As a profession, engineering is still in its infancy. It is not a closed profession, but is open to all who have attained specialized knowledge fitting them to carry out engineering enterprises successfully. He cited the engineer's tendency to stand in the background and expect those in need of his services to call upon him for advice as detrimental to his progress and the standing of the entire profession. That more engineers are not holding high executive positions is due, he believes, to the fact that engineers as a whole have not demonstrated to the public that the ability to manage men and industries is a fundamental characteristic of the profession.

At a joint session with the American Society of Refrigerating Engineers, W. H. McAdams and T. H. Frost, both of the Massachusetts Institute of Technology, presented a paper outlining the method of developing an equation which may be used by designers to predict the numerical value of the film coefficient of heat transfer between the clean pipe and the water which is flowing through it in turbulent motion. When used in connection with other data now available, this equation can be used to design apparatus such as condensers, feed-water heaters, and the like.

In all problems of heat transmission between water flowing inside pipes and any heating or cooling fluid outside, the heat flow is conditioned by three thermal resistances: that of the water film on the inner surface, that of the wall of the pipe, and that of the external fluid film. The first and last of these are determined by a number of factors independent for each of the two films. The total or overall thermal resistance is a complex variable. The individual film resistance rather than the overall is most desirable and the authors developed the following formula for the coefficient of heat transfer between the water and the pipe in which it is flowing in turbulent motion:

$$h = \frac{138}{D^{0.2}} \left(\frac{u}{Z} \right)^{0.8} \left(1 + \frac{50}{r} \right)$$

In which h = the film coefficient of heat transfer, between the inner wall of the pipe and the main body of water, the coefficient being expressed in B.t.u. per hr. per deg. F. per sq. ft. of inner wall. D = the inner diameter of the pipe expressed in inches. u = average velocity of water, in ft. per sec. Z_t = absolute viscosity of the water at the mean temperature of the film, Z_t being expressed in centipoises. r = ratio of actual length of pipe to its actual inside diameter, no units.

That the power department of a textile plant should be subdivided into two departments, one having for its duties the generation of steam and power, the other, supervision over the use of the products of the power plant, was the underlying thought in the paper by Henry M. Burke, of the Mt. Hope Finishing Co., entitled "A Steam-Loss Prevention Plan Operating in a Textile-Finishing Plant."

Duties of the power despatcher were outlined as: the inspection of all distribution lines, and conductors (except sprinkler system), and all valves, traps, appurtenances, and instruments thereon, for the purpose of preventing any losses of heat, light, and power in transmission; investigations and studies of the various manners in which steam, water and power are used for various purposes, with the aim of devising and recommending a more economical practice; and the recording of steam, power, and water consumption by various departments and preparation of reports and cost charges. He is given authority to schedule and specify all maintenance work on distribution lines and appliances, to supervise this work, to secure information, data and records from the heads of departments, and to inspect the actual practice of drawing and consuming heat, light, power and water in every department.

The power despatcher is held responsible to the plant engineer for uninterrupted and economical delivery of heat, light, power, and water distributed through conductors under his supervision, for accuracy of all records and accounts covering distribution and use of light, heat, power, and water, and for effective co-operation with all departments for the purpose of securing greatest practical economy in the use of heat, light, power and water.

By making this division of responsibilities of the power department and the installation of adequate instrument equipment it was possible to establish an economical operating practice and standardize conditions such as distribution of load among boilers, putting on and taking off load from boilers at the proper time, regulating draft according to fires and fires according to load, cleaning of heating surface by removal of soot and scale, and the method of firing coal-stoked boilers.

In the discussion of this paper, the need for cutting down the cost of production was the prevailing thought. Daniel M. Bates, of Day & Zimmerman, told of a plant where the boilers were failing to meet the requirements because of a peak load between 11 a. m. and 2 p. m. By co-operation with the mill, this load was distributed over the entire 24 hr. and thus delayed an increase in boiler capacity for 6 yr. He also pointed out the value of visiting the mill when it is not working as the wastes are then more in evidence.

Under the title "Factors in the Spontaneous Combustion of Coal," O. P. Hood, of the U. S. Bureau of Mines, called attention to the fundamentals to be observed in storing coal with the minimum danger of spontaneous combustion. When mined it is the freshly

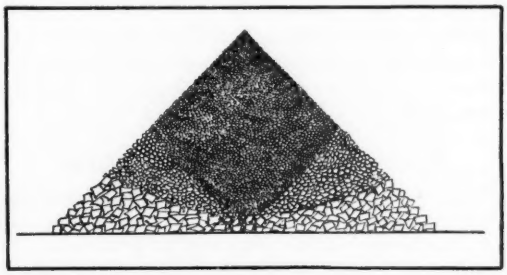


FIG. 1. DROPPING COAL CONTINUOUSLY IN ONE PLACE FORMS CONICAL PILE AND SEGREGATES LUMPS AND FINES

broken coal surface that experiences the greatest change and which finds its equilibrium most seriously upset.

Powdered coal sealed in a bottle with air gradually takes the oxygen of the air into combination, reducing the pressure in the container. Some CO_2 is formed, but not enough to account for all the oxygen. There is slow combustion and other reactions less definite and less stable. Some coals will take up as much as 6 per cent in weight in this process. Small amounts of heat may be generated in other ways. We may expect a fresh broken surface of coal to be chemically active in some way, seeking to establish equilibrium under new conditions. The amount of the activity will vary with the particular coal, the amount of surface, and the environment. It will be rapid at first, and finally be so slow as to pass out of our interest. These changes may develop heat; they may add to the weight, and may finally reduce the weight by inducing combustion.

In storing coal, there must be no heat added to the coal from outside sources, such as hot walls, steam pipes, oily waste, etc. If coal can be got into the pile in pieces that will stay on a ¾-in. bar screen there will be no heating. The pile may be of any height provided there be no fines. Where mixed lump and fine coal must be stored, the pile should be so built as to avoid segregation of sizes. The production of fresh surfaces by breakage just before going into storage, predisposes to spontaneous combustion unless the crushing is so fine and the packing of the pile so dense as largely to exclude air circulation.

Coal should be so stored that its temperature conditions can be daily inspected. When any portion reaches a temperature of 140 to 150 deg. F., there is a high probability that within a few days or weeks a destructive temperature will be reached. If the temperature reaches 160 or 180 deg. F., there is almost a certainty that a

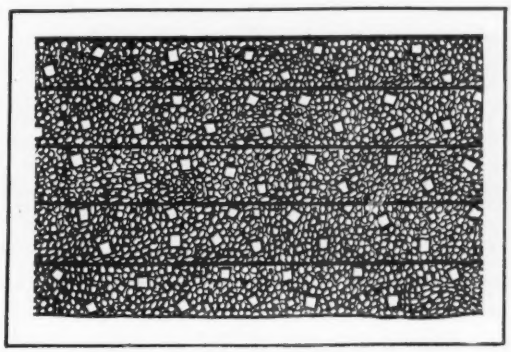


FIG. 2. WHEN PILED IN HORIZONTAL LAYERS UNIFORMITY OF LUMPS AND FINE COAL EXISTS

destructive temperature will be reached. The best remedy is to move the warm coal so that it may cool. The means for quick removal, rather than any tendency to heating, determines the desirable size of pile and its height.

Coal-Storage Systems

H. E. Birch, of the R. H. Beaumont Co., and H. V. Coes, of Ford, Bacon & Davis, stated in their paper on Coal-Storage Systems that the amount of coal storage desirable depends on the factors of distance from source of supply, adequacy of rail or water connections, the seriousness of a possible shutdown, the probable future growth of the plant, and perhaps other local considerations which must be weighed separately for the particular plant in question.

Due to the liability of spontaneous combustion, coal should be stored in such a manner as to prevent its occurrence, and a storage system should be selected that can be utilized easily and rapidly to fight a fire which may occur through failure to store coal properly.

Many devices store coal by the cone method shown in Fig. 1. The pile thus formed consists of lumps at the base and fines above. Sufficient air is admitted at the base to start and maintain oxidation, while the blanket of fines prevents the escape of the generated heat. If coal is discharged from an overhead telpher, trestle, or fixed conveyor the same condition obtains for the full length of the pile. By piling the coal in horizontal layers, as indicated in Fig. 2, the interstices are filled with fines producing a more or less homogeneous mass in which the oxidation is prevented or retarded due to lack of air.

Advantages and limitations of various devices and systems were dealt with. The problem of unloading railroad cars was considered complex, since the types of cars vary considerably and the coal may be frozen solid in the car. For unloading boats, the steeple tower, mast and gaff, stiff-leg-derrick, locomotive crane, bridge

tramway, man trolley and monorail telpher all came in for discussion.

In regard to the locomotive crane and trestle when used for storing coal, it was stated that we cannot afford the expense of covering the entire storage area with a trestle and must therefore use it merely as an auxiliary to the complete system, primarily as an unloading means and as a means of forming a long pile, and so eliminating the necessity of the crane traveling with each bucket load. The illustration, Fig. 3, gives the cross-sectional dimensions of the coal pile, both with the crane tracks left open and covered, although the authors do not recommend covering the crane tracks due to the difficulty of handling the coal in case of fire.

To be ideal, the storage plant must be so designed that it can be easily extended; must meet the operating, maintenance and interest formula; be capable of unloading boats and railroad cars rapidly and economically; will store coal so as to prevent spontaneous combustion; will reclaim coal economically and can fight fire

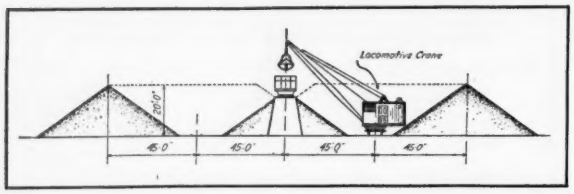


FIG. 3. WITH A LOCOMOTIVE CRANE AND TRESTLE THE STORAGE PILE SHOULD NOT COVER THE TRACKS

effectively; should be mobile enough readily to adapt itself to an entirely new location; and trackage facilities should be laid out to handle the coal traffic without interrupting the general plant traffic.

Discussion at Coal Storage Session

In discussing papers presented at the coal storage session, J. E. Davenport stated that coal storage is the solution to the railroad car shortage problem, one-third of the freight handled being coal. With production peaks eliminated as in 1923 present railway capacity is sufficient for all present needs. C. G. Spencer advocated buying coal of a quality that does not ignite when stored. He questioned the statement that depth of coal pile does not influence liability to spontaneous combustion and believes that selection of storage system depends upon the character of coal handled. To provide storage equipment for too far into future may lead to needless expense when plant conditions change such as to use of powdered coal.

George A. Orrok declared that although it costs from 50c to $1 a ton to store coal, an industry should make it a practice first to insure against a shutdown, and second as protection against high prices of spot coal. His experience indicates that 15 to 20 per cent of value of coal is lost by deterioration during storage.

Storing coal is an absolute necessity with public utilities was the statement by John W. Lieb. It costs money but 60 to 90 days' supply should be on hand for use during periods of stress. He does not find chemical analysis a guide in selecting coal for storage and expects fires to break out occasionally.

Coal can be stored so as to prevent spontaneous combustion, said Charles R. Richards, and tests show that the loss in heat value is negligible. Fires must be handled differently when burning stored and freshly mined coal.

Applying Mathematics to Boiler-Plant Performance

Under the title Boiler-Plant Economics, N. E. Funk and Farley C. Ralston, both of the Philadelphia Electric Co., presented a paper showing how a formula which they developed may be applied to boiler-plant performance, and also discussing the question of operating at minimum cost. It would seem to the authors that the operation of boiler plants has received entirely too little attention of the mathematical kind. Most of the work done on the subject has been developed graphically; but this method of procedure, while very valuable, is less desirable for many purposes than direct calculation by means of analytical formulas.

For logical treatment, the paper was divided into three parts; one is devoted to the development of a rational formula for expressing boiler-plant performance, and to the determination, for boiler-efficiency guarantees and tests, of the values which the constants in the formula take for any given boiler. In the second part, the boiler formula is applied in solving the problems of operation at maximum boiler efficiency, at maximum boiler-room efficiency for a given load, and at maximum daily boiler-room efficiency. In the third part, the problem of operation at minimum total cost, including investment, operating and maintenance charges, is considered. An abstract of this paper will be given in a future issue.

Discussions of Steam Power Papers

There is a distinct gain in boiler efficiency by the use of preheated air due to better combustion conditions which is over and above that due to heating air alone, declared A. G. Christie; in this case it is 0.65 per cent. What is needed today is a simple, cheap form of air preheater that will be permanent and will remain clean and that will give efficiencies of the same order as the boiler efficiency.

That it will not do to draw too positive conclusions from this particular performance, was the thought of R. Sanford Riley. The fusing temperature of ash in the coal is not mentioned but we must assume that it is relatively high. Serious trouble has been reported from France where the preheating was only 72 deg. F.

Heat absorbing efficiency of the heating surface of the boiler and the superheater is not increased by preheated air, said Leo Loeb, but the gains recorded are due first to increased furnace efficiency and secondly by increased absorption of radiant heat.

B. N. Broido suggested reheating steam in a separately fired superheater located in the turbine room, also reheating by live steam from the boiler.

Small power plants would decrease plant investment by substituting motor driven auxiliaries for many of the non-condensing turbines and heating feed water by steam bled from the main turbine, was the opinion expressed by S. A. Moss.

Walter M. Keenan suggested that turbine manufacturers furnish power plant designers as part of their proposals, Mollier or similar charts showing the steam conditions throughout the machine.

In the design of a plant, said O. F. Jungren, we must study where the cost of fuel will go during the life of the plant and how the load factor is going to be. A

high load factor warrants greater expenditure for efficiency.

That preheated air will cause more rapid depreciation of the furnace walls was brought out in the discussion but refractory manufacturers report progress that will keep up in advances in furnace temperatures.

Because accompanying preheated air furnace temperatures are reached that fuse coal ashes, A. A. Adler thought its use would be limited to pulverized coal and oil.

H. C. Heaton believes the predictions as to economy of high-pressure steam will be borne out but cautions that the critical temperature of the materials now in use is about 750 deg. F. above 400 lb. pressure serious difficulties enter for 1200 lb. boilers now under construction forged steel drums are used.

Circulatory heat was the term used by George A. Orrok in describing the heat that can be recovered from the flue gases and the condenser. The turbine of today probably is not the most efficient for the pressures and temperatures under discussion and when the properties of steam are known up to 1500 lb. more efficient designs will be possible.

Bleeding Turbines to Heat Feed Water

E. H. Brown and N. K. Drewry, both of the Allis-Chalmers Manufacturing Co., pointed out in their paper on "The Economy Characteristics of Stage Feed-Water Heating by Extraction," the influence of each possible factor in the computations, presented a practical method for computing extraction cycles and dealt with various factors affecting turbine room economy.

Turbine principles as stated by the authors are summarized as follows: (a) total steam passed is directly proportional to inlet pressure; (b) inlet pressure varies according to a straight-line-with-intercept relation to load; (c) extraction nozzle pressures bear the same relation to load as does the inlet pressure; (d) extraction nozzle temperatures varying with nozzle pressures also bear the same relation to load; (e) nozzle pressures and temperatures decrease proportionately with the amount extracted for given loads by amounts dependent upon the position of the nozzle.

It is stated, as the opinion of the authors, that the use of combustion air economizers, adoption of the unit system, improvement in the reliability of auxiliaries, and the growing demand for simplicity in power plant construction, all point towards a steady adoption of the stage extraction method of heating feed water.

That air preheaters can effectively lessen the economical rating of feed-water economizers, thus allowing greater benefits of extraction feed water-heating to accrue by heating to higher temperatures than the turbine room, seem probable. Mechanically and thermally successful air heaters have been tried and tested with credible results decreasing chimney temperatures to exceedingly low amount while carrying part of economizer heat loads. Increasing the turbine-room feed-water temperature to 300 deg. F. from the present practice of 210 deg. F. is therefore not improbable. The change would decrease economizer corrosion, increase boiler rating and allow a gain in overall plant economy. The increase in feed-water temperature mentioned above would additionally improve turbine-room heat consumption 1.25 per cent if a two-heater system were in use; and 1.95 per cent for a three-heater system.

Reheating in Central Stations

Under the above title, W. J. Wohlenberg, of Yale University, contributed a paper in which he discusses the influence of amounts of energy added in reheating, number of reheating stations and points in expansion at which reheating should begin. A comparison was made of reheating, regenerative and combination cycles combining both reheating and bleeding stages, and it was shown that the cause of the influence on the internal machine efficiency thereby, reheating properly applied may lead to higher efficiency than bleeding.

It was furthermore shown that the combination cycles give promise of realization in practice of appreciably higher efficiencies than would be the case of the other cycles investigated. The analysis also indicates that for a limiting steam temperature of 750 deg. F. the maximum efficiency is attained at initial pressures varying between 600 and 900 lb., the exact optimum depending upon cycle fuel and steam generator used; and that if pressures are increased beyond these points there will result an actual reduction in overall efficiency.

The paper goes into a study of the temperature entropy diagrams for stage reheating and bleeding cycles and then takes up a discussion of the combined cycles, dealing with both one stage and two stage reheating and bleeding performance.

An abstract of this paper will be given in a future issue of this magazine.

Possible Improvements in the Steam Plant

Ernest L. Robinson, of the General Electric Co., presented a paper discussing the maximum attainable efficiency of the ideal heat engine, and circumstances which limit it in order to point out the margins available for improvement along different lines. The arrangement of the circulating systems is considered and after noting the ideal arrangement practical layouts are taken up. The advantages of the mercury turbine and the steam extraction cycle are emphasized and the great increase in capacity rating which has been made possible by purely thermodynamic improvements is also noted.

If the ideal heat engine could convert all the heat into work, it would be necessary to reject the working substance devoid of all heat content, that is, at absolute zero. If, at any time, energy is reclaimed from the working substance without other heat exchange, its availability to do work has been decreased a corresponding amount, that is, its absolute temperature has been changed. In fact, this idea is the whole basis for the definition of the thermodynamic scale of temperature. Since the temperature of a condenser can hardly be lowered to absolute zero it is necessary for heat engines in general to do work even under ideal circumstances at an efficiency far less than unity.

This efficiency is represented by the actual temperature drop of the working substance divided by what the drop might be if all heat were abstracted, that is, the initial absolute temperature. This is the Carnot cycle efficiency and also the efficiency of any reversible engine.

It is the author's idea that on the supposition of the perfect transfer of all the heat of the flue gases to the entering air and fuel there would be no thermodynamic loss in the furnace and if in addition an isothermal transfer of the heat of combustion to the boiler should occur, then the boiler could be considered as ideal.

Metropolitan Section A. S. M. E. Discusses Powdered Coal

THAT the subject of pulverized coal is one of live general interest was apparent from the large attendance when H. D. Savage, of the Combustion Engineering Corporation, and F. A. Scheffler, of the Fuller Engineering Co., talked on that subject before the Metropolitan Section of the A. S. M. E. on Nov. 20. Both speakers had provided lantern slides showing types of equipment and interesting installations, as well as test data, and the discussion following the presentation of the papers brought out suggestions and details as to design and operating results. Strangely enough, however, there seemed to be no stoker men on hand to discuss the subject from their point of view, so that on this occasion the pros may be said to have carried off the honors undisputed.

Mr. Scheffler pointed out the rapid growth of use of pulverized coal, saying that a year ago 15,000,000 T. were being employed annually in this country and Canada, but that when construction work undertaken this year is completed the total tonnage will be increased to 17,500,000 T., or over 15 per cent. As evidence of the general interest in this type of fuel burning, he listed a wide variety of industries in which successful application of it has been made.

Higher Operating Temperatures Prophesied

Good practice in determining furnace volume is still based, said the speaker, on the factor of 7 to 10 ft. per sec. for speed of gases passing through the furnace. The corresponding furnace volume is about 1¾ to 2 cu. ft. per boiler horsepower developed continuously. It is believed, however, that a special brick will soon be developed at a moderate cost which will permit continuous operation at 3400 deg. F. and reduction of furnace volume by 20 per cent to 25 per cent.

Safety of operation is insured with properly designed apparatus, and comprehensive rules have been prepared through the co-operation of the manufacturers with the National Board of Fire Underwriters, and are about to be adopted by the latter. The leading manufacturers have now been working with this equipment for the past 20 yr., so that the types have become standardized. It is possible to distribute the pulverized coal for a distance of over a mile from the pulverizing mills to the point of utilization at a reasonably low cost, and this is being done, as was shown by slides of such an installation.

Mr. Scheffler said that reliable firms, in giving out information on the cost of pulverizing, let it be known that they include cost of crushing, drying, repairs to entire equipment, distribution to boiler bins and furnaces, and air supply. In comparison with stoker operating costs, he pointed out that the stoker systems must be charged with such items as preliminary crushing, conveying coal to bunkers, operating of stoker mechanism and of fans for air supply, and repairs to all equipment concerned. With these allowances, he showed that the difference in operating costs on a per ton basis would be reduced to a few cents, and presented tables to substantiate this point.

Mr. Savage's paper dealt largely with statements from various sources showing the success which had been met with pulverized coal installations with various conditions and kinds of coal. He, too, laid stress upon the increasing popularity of the system, and enumerated the advantages claimed for it. If such factors as reduced heating surface of boilers required for the same plant capacity be taken into consideration, in figuring first costs for plants designed for pulverized coal as against stokers, he said that the former would show little if any excess. He also asserted that the operating costs are no higher than in the case of stoker installations, for the larger plants.

Operation at Cahokia Successful

Opening the discussion, Col. Junkersfeld, of McClellan & Junkersfeld, said that the installation at Cahokia had been running about one month, but they have great confidence in its success and intend to make further installations of the same character. Mr. Keith, who has had charge of operations at Cahokia for this firm, gave some details concerning conditions there. He said that in starting they ran up to 300 lb. pressure in 2 hr., then cut down and examined the setting for cracks, but found none and started again. So far, the maximum rating attained has been 235 per cent, as they are going slowly in this matter. The boilers have 10 burners, and they have about 40 per cent more to go on the variable speed motors before reaching the ultimate capacity. Coal of 5 per cent moisture, 13 per cent ash, and 11,500 B.t.u. is used. It is ground in 6-T. mills; but as 2000 T. per day must be provided for, the size of mills will be increased to 15 T. No trouble has been encountered from caking of coal. Slag on side walls has been eliminated by tipping the side burners toward the center. They have not operated a boiler over two weeks at a time, and give a careful examination on taking a boiler off the line. From experience with both kinds of installation, Mr. Keith said that they had had no more trouble than would be met with new stoker equipment.

Mr. Jacoby, of the J. G. White Corporation, gave some account of a small installation in Columbia, S. A., where extremely simple pulverizing equipment is employed, and only native labor is available. On account of the great cost of importation, furnace brick made in the vicinity of the installation had to be used. The success of the plant has been so marked as to bring many inquiries from the same country. On the general subject, this speaker said that the economic and business side of the pulverized coal question is fairly well understood, and expressed the opinion that this equipment had found a field of its own which it would retain. He believed that the tendency toward greater furnace volumes should be checked.

Prof. Chrystie, of Johns Hopkins University, spoke of the possibility for further development through utilization of radiant heat, disposing of heat around the side walls. If we put too much heating surface at the side walls, however, the resultant cooling effect will lower the efficiency. The remedy may lie in preheating the air and aiming at a Bunsen burner effect. As an efficiency of 80 per cent is already expected from powdered coal, the field for additional improvement is limited.

Mr. Frost, of the Public Service Electric Co., said that they had decided in favor of stokers for their latest installation.

A comparison of evaporation from bottom row of boiler tubes and from side wall water screens was made

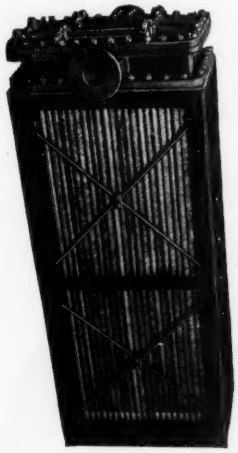
by Mr. Piggott, of Stevens & Wood. From the former he said we can develop 80 lb. per sq. ft. of surface, while the latter will give from 40 lb. per sq. ft. up to possibly as high a figure as the bottom row. He agreed with Prof. Chrystie on the desirability of preheating, but as furnace volumes are relatively large compared with cooling surface, he doubted if water screens would depress the temperature of flame over 100 deg.

On this question of flame temperature, G. A. Orrok doubted if they could be reduced by any type of furnace wall. He believes that they would be in the neighborhood of 3200-3500-3600 deg. F. What counts, however, is the average temperature of the whole furnace space, not the flame temperature alone.

Condensate Used for Cooling Generator

TURBINE condensate, passing from the surface condenser to feed heating apparatus is used as a medium for cooling generator air in a new type of cooler which has recently been developed by the Griscom-Russell Co., of New York City.

This air cooler, known as the U-fin, is designed for installation under the main station generator and is



HEAT EXTRACTED FROM THE GENERATOR AIR RAISES CONDENSATE TEMPERATURE 10 DEG. F.

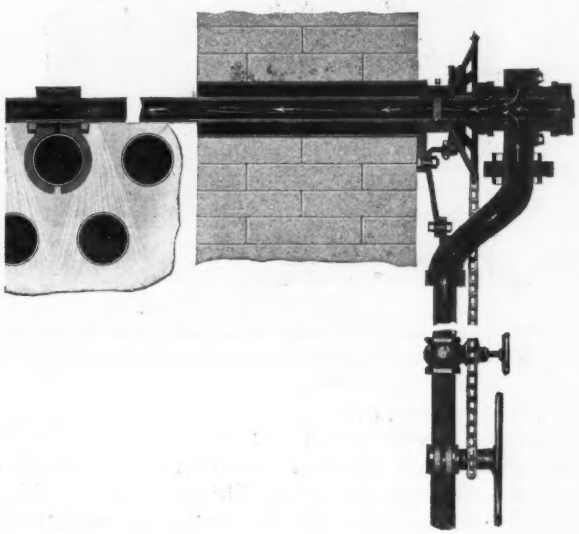
enclosed in the duct work. Its purpose is to cool the ventilating air discharge from the generator windings. In the past this generator air, which must be forced through the windings in order to keep them properly cooled, has been discharged into the generator room and this heat is not only lost, but it has been a source of discomfort to the operating force. The U-fin cooler utilizes the heat which is extracted from the air, the temperature of the turbine condensate on its way to the hot-well is increased from 8 to 10 deg. F. This cooler is of the surface type and the cooling of the air is, therefore, accomplished in a closed system which permits the use of the same air over and over again and, in addition, greatly reduces the fire hazard, because if a fire should occur in the generator windings, it would be quickly extinguished due to the lack of sufficient air to support combustion for any length of time.

The U-fin air cooler is a compact, rigid unit, the sides of which are encased in sheet iron. The cooling surface is formed of rigid U-fin tubes of ⅝-in. Admiralty metal with rectangular brass fins. Turbine condensate passes through the tubes and the generator ventilating air is forced through the cooler casing and is cooled by its contact with the fins attached to the tubes.

It is stated that coolers sufficient for 500,000-kw. generator capacity are on order.

Soot Blower Has Balanced Head

ELIMINATION of end thrust by the use of a balanced head is the feature of a new soot blower now being built by the Marion Machine, Foundry and Supply Co., of Marion, Ind. As will be noted from the accompanying cross-section, the steam enters a steam chest and then through ports into a swivel tube which, combined, gives several times the area of the soot blower element.



END THRUST IS ELIMINATED BY ADMITTING STEAM AT SIDES

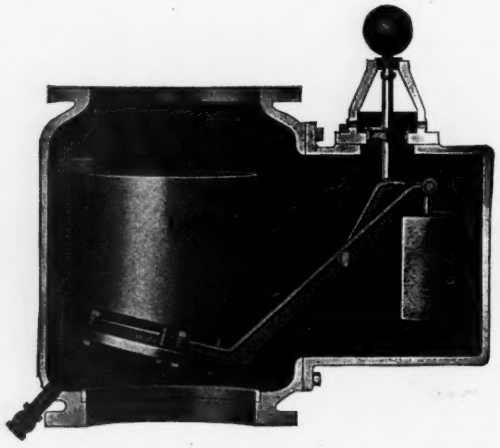
The swivel tube passes through the steam chest, being closed on the outer end, and is mounted between two stuffing-boxes.

Since the steam surrounds this tube and enters at the side through the ports, all thrust is on the inside of the tube and not on the steam head. There is, therefore, no end thrust or tendency to upset the packing. Furthermore there is no strain on the packing except the steam pressure.

At the end of the element, there is a welded coupling. The gear wheel is mounted on this coupling and secured by two set screws eliminating any possibility of the nozzles getting out of alinement with the indicator on the gear wheel. The swivel tube is then securely screwed into the coupling and is free from any strain that may be thrown on the sheave wheel due to warped elements or binding and it is impossible for this swivel tube to back out of the coupling after it is once bolted in the head or held in position by the end plates.

Vacuum Breaker Operates by Weight of Water

IN A combination vacuum breaker and non-return valve recently brought out by the Morton Vacuum Breaker Co., of Canton, Mass., the breaker is actuated by the weight of the water. The breaker forms a portion of the vertical exhaust pipe and the exhaust steam passes up around the bucket which is larger than the inside diameter of pipe. The bucket rests on two knife-edge bearings, and is held in an upward position against a stop by a counter weight. When the condenser floods back, the condensing water which would cause an accident, enters the bucket, which when about one-quarter full, will travel down opening the air valve, destroying the vacuum, and the valve plate below the bucket closes



WHEN BUCKET IS ONE-FOURTH FULL, AIR VALVE IS OPENED AND PLATE VALVE CLOSES AS A NON-RETURN VALVE

as a non-return valve. After valve has operated, the drain cock may be opened, and the water will all drain out. The bucket will then rise to its upward position and the air valve will close. The condensation which collects in the bucket runs out through the small holes at the bottom.

This type of valve is used where the exhaust pipe rises up to the jet or barometric condenser, or in a vertical dry air pipe where a reciprocating pump is used on a surface or jet condenser.

If the non-return valve is used to protect a dry air pump on a surface or jet condenser, the air valve may be omitted causing no interruption of vacuum. The dry air pump will only be out of service until line is drained of water.

Power Plant at Muscle Shoals May Replace Gorgas Unit

TO COMPLY with the Ford offer for the Muscle Shoals, Ala., plant, the construction of a new auxiliary steam power plant to replace the Gorgas plant disposed of recently to the Alabama Power Co. is provided for in a bill prepared by Representative Madden, Illinois, for introduction upon assembling of the next Congress.

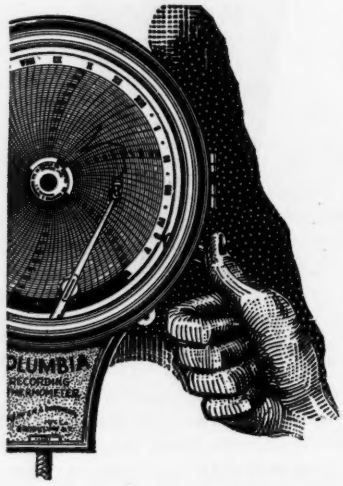
This is a revival of the legislation which was not passed on in the last Congress, with the addition of a new section designed to meet conditions resulting from the sale of the Gorgas steam power plant.

The new section as proposed by Representative Madden would have the government, through the War Department, substitute an auxiliary steam power plant with a capacity of 40,000 hp. The proposal is for construction of the plant directly by the government or its construction under contract by Henry Ford or his corporation. The bill would also provide for acquisition by the War Department, through condemnation or otherwise, a site for the steam power plant on the Black Warrior River near lock No. 17 with a strip of land connecting to dam No. 2.

The statement of Representative Madden, that, "The manufacture of fertilizer at Muscle Shoals is the vital consideration. Almost anyone could develop the power, which is limited to 100,000 primary horsepower, but manufacture of fertilizer would require an immediate investment of from $50,000,000 to $60,000,000 and Mr. Ford is the only bidder whose offer is based on the plan to manufacture fertilizer so badly needed by the agricultural interests of the South as well as other parts of the country," explains most thoroughly the attitude of the government in attempting disposal of Muscle Shoals to parties with funds to finance operation of the tremendous plant.

Patent Infringement

IT HAS recently been called to our attention that the use of a recording gage for a time punch, as described in our Oct. 1 issue on page 1002 by John H. Schalek, is an



PATENT TIME PUNCH FEATURE APPLIED TO RECORDING GAGE

infringement on patents held by the American Schaeffer & Budenberg Corp. These patents cover the apparatus shown in the accompanying figure, the "Time Punch" as made by this company, and also the method explained by Mr. Schalek.

TURBINE driven boiler feed pumps are considered easier to control than motor driven pumps due to the fact that the excess pressure regulator is generally considered to be better developed than any automatic means at present available for controlling motor driven pumps.

New El Dorado Hydroelectric Plant will be Finished Soon

CONSTRUCTION of the new El Dorado hydroelectric plant for Western States Gas & Electric Co. is proceeding ahead of schedule. Planned originally to be in operation early in 1924, it now appears that, barring unforeseen circumstances, the wheels will be turning before Christmas of this year. And thus will be added 27,000 hp. to the wealth of the hydroelectric resources of California.

During the greater part of the year, from 1250 to 1500 men have been employed on this work, and during the summer months work proceeded 24 hr. a day on parts of the development. This was made necessary by the short working season, due to early snowfall. The Twin Lakes storage reservoir, which impounds 20,000 acre feet of water, has been completed. The construction of the 23-mi. canal, of which 5 mi. have been concrete lined, is virtually complete and will be finished in a few weeks. The forebay, penstock, power house building and transmission lines are now complete and the installation of machinery is now about 75 per cent accomplished.

Construction of El Dorado power plant is of great importance to the Western States Gas & Electric Co. and the territory served by that company. The development work has extended over a period of several years, during which time the demand for power in the territory served by the company has so far exceeded the amount of power available that it was necessary for the Western States Gas & Electric Co. to purchase power outside its own developed resources. With the completion of this plant, the company will be able to substitute generated power for power now purchased, and it will be able more adequately to meet the growing demand for electric energy in Stockton and Richmond and the surrounding territory.

Western States Gas & Electric Co., in prosecuting its development of the hydroelectric resources on the south fork of the American River, has had in mind not only this first unit of 27,000 hp., but additional units which will be constructed as demand for power increases.

Secretary Work in Accord with Plans for Reorganization

THE AMERICAN Federated Engineering Societies plan for reorganization of certain bureaus of the Interior Department is in agreement with Secretary Work's views for re-arrangement of the departments along technical lines. Lawrence W. Wallace, executive secretary of the engineering organization, after conferring with Secretary Work, announces that it is found that the plan of the engineers agrees in detail with the Brown plan for reorganizing, technically, the bureaus of the Interior Department.

These plans contemplate establishment of a bureau of public works to be under the administration of an engineer as an assistant secretary. It is understood that the engineering societies will press the adoption of the Brown plan at the national conference held in Washington at the first of the year.

The president of the American Engineering Council, it is stated, has been authorized by the executive board to appoint a special committee to draft for introduction into Congress a bill carrying in effect that portion of the reorganization plan which deals with the Interior Department. The transfer of the bureau of mines and the patent office are contained in the Brown plan and it is considered most desirable to head these with the best engineering talent.

Syndicate to Seek Boulder Canyon Rights

IMMEDIATE construction of the Boulder Canyon project, at a cost of $150,000,000 to be financed by a New York syndicate of banks will be proposed to Congress on Jan. 10, 1924, according to W. G. Clarke, chairman of the corporation's board of engineers, who was in San Francisco recently.

An investigation which has been conducted for several months and which is just now being completed has given up data from which it has been possible to ascertain the cost, the dimensions of the dam and the fact that it is entirely feasible.

It proposed to impound the silt in the Colorado, preventing further filling of the channel; to control the flood water in a reservoir; to build an immense dam, and construct a power plant able to generate 1,000,000 hp. These improvements to cost $150,000,000 will be turned over to the government on completion, upon which the government will give the corporation a permit to distribute the power under entire government regulation.

News Notes

JULIAN C. SMITH, general manager of the Shawinigan Water and Power Co., one of the largest power companies in Canada, delivered the second lecture in the Aldred Course at the Massachuetts Institute of Technology, at Cambridge, Mass., on Nov. 23 to an audience of faculty members, graduate students, and seniors.

H. L. UNLAND, for 13 yr. engineer with the General Electric Co., 11 yr. of which were spent in the Power and Mining Department, left that concern Dec. 1, to take the position of electrical engineer with the Victor Talking Machine Co., at Camden, N. J. Mr. Unland is a graduate of the Engineering College of the University of Nebraska, class of 1910.

DWIGHT P. ROBINSON & Co., of New York, has opened an Atlanta office in the Healey building, with W. Rawson Collier in charge. Mr. Collier was for many years with the Georgia Railway & Power Co. He joined the Dwight P. Robinson forces several months ago, coming to them from the Poughkeepsie Gas & Electric Co. He is a graduate of the Massachusetts Institute of Technology, and has been identified with the work of the National Electric Light Association, the American Institute of Electrical Engineers, the American Gas Association and the Illuminating Engineering Society.

NORTHERN STATES POWER Co. is in process of acquiring control of the Wisconsin-Minnesota Light and Power Co. by purchase of common stock. The completion of the transaction will give the Northern States Power Co. a generating capacity of 226,763 kw. and transmission system of 6396 mi. of line covering terri-

tory from La Crosse to Rice Lake in Wisconsin, from northern Iowa to Minneapolis and from Edgar, Wis., to Clarkfield, Minn. There are also five detached small systems centering about Grand Forks, N. D.; Fargo, N. D.; Sioux Falls, S. D.; Galena, Ill., and Ottumwa, Ia. The system will be managed by the Byllesby Engineering and Management Corp'n.

PLANS HAVE been prepared by the federal government for an addition to the power plant at the government station at Melville, R. I., and tanks are being installed with 80,000-bbl. capacity for fuel oil.

MCCLAVE-BROOKS Co., Scranton, Pa., announces the appointment of Byron L. Ashdown, 1236 Syndicate Trust Bldg., St. Louis, as sales representative in St. Louis and surrounding territory in Missouri and western Illinois.

L. R. BROWN has been appointed manager of the transformer division of the new central station department of the General Electric Co., according to a recent announcement. He was one of the company's first transformer specialists and has a wide personal acquaintance in the central station industry. Mr. Brown is a graduate of the University of Wisconsin, class of 1903.

COMBINATION has been effected of the Westinghouse Electric and Mitsubishi and the Takata electric interests to form the Mitsubishi Electric Co. which will manufacture and sell electric equipment in Japan. Capital will be $7,500,000, American interests supplanting German in the Takata group. Plants exist at Kobe, at Nagasaki and at Tagoya, the last named to be enlarged at once. Patent licenses will be exchanged between the Westinghouse Electric & Mfg. Co. and the Japanese interests.

Catalog Notes

TWO NEW CATALOGS have been issued by the W. A. Jones Foundry & Machine Co., of Chicago, one of which covers pulleys and the other friction clutches.

DEAN CATALOG No. 10, describing the Dean boiler room equipment, grates, ash disposal apparatus and furnace linings, is now being distributed by Washburn & Granger, Inc., of New York City.

RELIANCE ELECTRIC & ENGINEERING Co., of Cleveland, Ohio, has recently issued an interesting booklet, "Electric Motors, How to Choose and Use Them." It is simply written and contains some valuable data regarding the selection of motors and method of installation.

EVERYTHING in Insulation is the title of a 160-page catalog published by the Mitchell-Rand Mfg. Co., of New York City. The catalog takes up such subjects as asbestos, insulating compounds, mica, insulating paints, papers, tapes, twines, varnishes, etc., and gives information on the production and use of such insulating materials.

AMES UNA-FLOW ENGINES are described in Bulletin 1924, which has just been published by the Ames Iron Works, of Oswego, N. Y. In this bulletin the difference between piston valve engines, single cylinder Corliss engines, compound cylinder Corliss engines, and the Una-Flow two-valve type of engine is presented in diagrammatic form. There is also a discussion of the early history of the Una-Flow engine and an explanation of the Una-Flow cycle of operation together with the thermal advantages as compared with the counter flow cycle. Construction and details of the Ames design are also described and illustrated.

IN A BULLETIN entitled "Boiler Trimmings" the Wright Austin Co. of Detroit, Mich., have described its line of safety alarm water columns, automatic pump governors and boiler feed regulators.

FULTON-DIESEL ENGINES are described in catalog No. 805 now being distributed by the Fulton Iron Works, of St. Louis. In addition to illustrations showing actual installations in the field, there are also photographs of the engine in various stages of erection and a detailed discussion of the construction features of this line of engines.

"IMPERIAL" COMMUTATOR STONES are covered in a folder which is now being distributed by the Martindale Electric Co., 11745 Detroit Ave., Cleveland. The advantages in using commutator stones for cleaning and truing up commutators are discussed and the various types of stones, together with the handles which may be used, are illustrated.

"SEVEN WONDERS OF WROUGHT PIPE" is a booklet issued by the National Tube Co., of Pittsburgh, in which some rather interesting and peculiar accidents are described in which wrought iron pipe has been forced to show its great ductility. These accidents have occurred in the oil fields and have been due either to dropping of casing pipe when it was hanging in a well or else due to pipe being blown out of a well.

CELITE PRODUCTS Co., of Chicago, is distributing engineering data sheets in loose leaf form on its heat insulation products. These sheets are for use in standard note books and include charts, showing the comparative conductivities of Sil-O-Cel brick, red brick and refractories as well as heat losses through various wall constructions, insulated and uninsulated, and drawings showing methods of insulating such equipment as boilers, furnaces, ovens, etc.

"MECHANICAL RESULTS FROM HAND STOKERS" is the subject of a bulletin published by the McClave-Brooks Co., of Scranton, Pa. It describes in detail the design features of the McClave hand stoker and discusses in a general way the application of this method of firing. Another bulletin in the same series, which is being issued by this organization for binding in one cover, is the McClave Hopper Feed Hand Stoker. This stoker is also described and illustrated.

TWO NEW CATALOGS have recently been issued by the C. J. Tagliabue Mfg. Co., of Brooklyn, N. Y., one of which covers oil testing instruments and the other power plant instruments. Catalog No. 699 describes oil testing instruments such as calorimeters, Saybolt viscosimeters and other devices used in testing fuel oils, lubricating oils and illuminating oils. Catalog No. 640A takes up CO_2 and CO recorders, recording thermometers, thermometers, vacuum gages, barometers, and other instruments which are built by this organization.